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ELEMENTS OF
HEAT-POWER ENGINEERING

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BY

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PART I. Thermodynamics and Prime Movers. Third Edition of First Part of Hirshfeld and Barnard's "Elements of Heat-Power Engineering," completely rewritten. 493 pages. 6 by 9. 251 figs. 7 plates.

PART II. Steam-generating Apparatus and Prime Movers, Fuels, Combustion, and Heat Transmission. Third Edition of Second Part of Hirshfeld and Barnard's "Elements of Heat-Power Engineering," completely rewritten. 871 pages. 6 by 9. 828 figs.

PART III. Auxiliary Equipment, Plant Ensemble, Air Conditioning, and Refrigeration. Third Edition of part of Hirshfeld and Barnard's "Elements of Heat-Power Engineering," completely rewritten. 415 pages. 6 by 9. 209 figs.

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PART III

AUXILIARY EQUIPMENT, PLANT ENSEMBLE,
AIR CONDITIONING, AND REFRIGERATION

THIRD EDITION

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HIRSHFELD AND BARNARD'S

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PREFACE TO PART III

THIS volume supplements the work in Part I on thermodynamics, and that in Part II pertaining to fuels, combustion, heat transmission, steam-generating equipment, and prime movers.

The text contained herein considers the fundamentals of the following subjects: flow of fluids through pipes, the draft system, the condenser system, feedwater heaters and related apparatus, boiler-water conditioning, properties of mixtures of air and water vapor, cooling ponds and towers, steam plant ensemble, economics and cost data, air conditioning, and refrigeration. The treatment is such that most of the chapters may be studied in any sequence desired, and various parts may be omitted in short courses. Numerous references are given so that the original sources of information can be consulted for further investigation. Many questions and problems are provided for each chapter.

The authors gratefully acknowledge their indebtedness to Professor Charles O. Mackey, who has prepared the chapters on the properties of mixtures of air and water vapor, on water cooling with atmospheric air, and on ventilation and air conditioning, and who has rendered valuable aid in many other ways; and to Professor Robert P. Kolb for his work on draft, boiler-water conditioning, feedwater heaters, and evaporators, and for other contributions. Special credit is due to Mr. Howard N. Fairchild for his able assistance in preparing many of the drawings and for rendering other faithful service. Thanks are extended to Professors Roy E. Clark and Warren H. Hook, and to Mr. Carl Shabtae, for valuable assistance of various kinds.

To the concerns who have furnished data, drawings, and other information, the authors express their sincere gratitude; in all cases an attempt has been made to give at the proper place in the text references to the sources of information drawn upon.

ITHACA, N. Y.
August 20, 1933.

CONTENTS OF PART III

CHAPTER XLII

	PAGE
THE FLOW OF FLUIDS THROUGH PIPES AND PASSAGES.....	781
498. General. 499. Resistance to the Flow of Fluids in "Short" and "Long" Lines. 500. Viscosity and Its Units. 501. Reynolds' Number. 502. Viscous Flow and the Friction Factor. 503. Turbulent Flow of Fluids in Pipes. 504. The Flow of Fluids through Bends, Fittings, and Valves. 505. Turbulent Flow of Fluids in Rough Closed Conduits. 506. Empirical Formulas. 507. Suitable Velocities. 508. Standards for Pipe and Fittings. 509. Steam Plant Piping.	

CHAPTER XLIII

THE DRAFT SYSTEM.....	837
510. Definitions of Draft. 511. Applications of Draft. 512. Natural Draft. 513. The Measurement of Draft. 514. Velocity, Static, and Total Pressures. 515. Mechanical Draft Fans. 516. Characteristics of Centrifugal Fans. 517. Propeller Fans. 518. Locomotive Draft. 519. Chimneys. 520. Air Ducts and Gas Breechings. 521. Removal of Flue Dust from the Draft System.	

CHAPTER XLIV

THE CONDENSER SYSTEM.....	870
522. General. 523. Measurement of Condenser Pressures. 524. Condenser Performance. 525. The Heat Content of Exhaust Steam. 526. Amount of Condensing Water Needed. 527. Jet or Direct-Contact Condensers—General. 528. Barometric Condensers. 529. Low-Level Jet Condensers. 530. Ejector Condensers. 531. Surface Condensers—General. 532. Typical Surface Condensers. 533. Tube and Hotwell Arrangements. 534. Surface Condenser Parts. 535. Condenser Connections and Supports. 536. Heat Transmission in Surface Condensers. 537. Design and Test Standards for Surface Condensers. 538. Economical Condenser Size, Arrangement, and Operation. 539. Condenser Auxiliaries—General. 540. Condensing-Water Supply Pumps. 541. Condensate Pumps; Removal Pumps. 542. Air Removal—General. 543. Air-Removal Apparatus. 544. Condenser Maintenance.	

CHAPTER XLV

FEEDWATER HEATERS, EVAPORATORS, PUMPS AND TRAPS.....	935
545. Introduction. 546. Advantages of Feedwater Heaters. 547. Classification of Feedwater Heaters. 548. Open-Type Feedwater Heaters.	

549. Closed Feedwater Heaters. 550. Energy Equations for Feedwater Heaters. 551. Heat Transfer in Closed Feedwater Heaters. 552. Adaptability of Open and Closed Heaters. 553. Feedwater Heating by Means of Special Equipment. 554. Uses and Advantages of Evaporators. 555. Classification of Evaporators. 556. Typical Arrangements of Evaporators. 557. Energy Equations for Evaporators. 558. Evaporation in Single and Multiple Effects. 559. Adaptability of Evaporators. 560. Types of Pumps. 561. Boiler Feed Pumps. 562. Drainage of Feedwater Heaters and Piping.

CHAPTER XLVI

BOILER WATER CONDITIONING.....	966
563. General Characteristics of Natural Waters and the Determination of Impurities. 564. Feedwater Make-Up. 565. Control of the Properties of Boiler Water. 566. Boiler Compounds. 567. Water Clarification. 568. Water Softening by Chemical Precipitation. 569. Commercial Chemical-Precipitation Softeners. 570. Zeolite Softeners. 571. Comparison of Zeolite and Other Softeners. 572. Phosphate Treatment. 573. Feedwater Treatment by Evaporation. 574. Foaming and Priming. 575. Control of Concentration in Boiler Water; Blowdown. 576. Deaeration. 577. Embrittlement of Boiler Steel. 578. Summary.	

CHAPTER XLVII

PROPERTIES OF MIXTURES OF AIR AND WATER VAPOR.....	990
579. General. 580. Principles and Definitions. 581. Properties of Mixtures of Air and Water Vapor. 582. Processes of Mixtures of Air and Water Vapor. 583. Psychrometric Charts.	

CHAPTER XLVIII

WATER COOLING WITH ATMOSPHERIC AIR.....	1005
584. General. 585. Heat Transfer in Water Cooling. 586. Cooling Ponds. 587. Spray Ponds. 588. Cooling Towers. 589. The Energy Balance for a Cooling Tower. 590. The Design of Cooling Towers. 591. The Selection of a Water-Cooling System.	

CHAPTER XLIX

THE STEAM POWER PLANT ENSEMBLE; LOCATION, ARRANGEMENT, ECONOMICS, COSTS.....	1020
592. General Factors Affecting Plant Location and Design. 593. Some Typical Steam Power Plants. 594. Combinations of Apparatus for Conserving Energy. 595. Regenerative Feedwater-Heating Systems. 596. Drainage of Regenerative Heaters. 597. Auxiliary Drives. 598. Conservation of Energy in Industrial Power Plants. 599. Analysis of Energy-Balance Systems. 600. Plant Economics. 601. Load Curves; Definitions. 602. Cost of Steam Power Plants. 603. Cost of Energy.	

CHAPTER L

	PAGE
VENTILATION, HEATING, COOLING, AND AIR CONDITIONING.....	1055
604. General. 605. Human Comfort and Air Environment. 606. Ventilation. 607. Heating Systems—General. 608. Hot-Water Heating Systems. 609. Steam Heating Systems. 610. Indirect Heating Systems. 611. Central and District Heating. 612. Unit or Room Coolers. 613. Air Conditioning. 614. Maximum Heating and Cooling Loads.	

CHAPTER LI

MECHANICAL REFRIGERATION.....	1087
615. Definition. 616. Basic Principles; the Carnot Cycle. 617. Gas and Vapor Refrigeration Cycles; General. 618. Air Refrigeration Cycles. 619. The Vapor Refrigeration Cycle. 620. Capacity, Rating, and Performance of Refrigerating Machines. 621. Relative Advantages of Various Refrigerants. 622. Compressors. 623. Refrigerant Condensers. 624. Precoolers and Accumulators. 625. Expansion Valves and Control Systems. 626. Evaporators. 627. Dry Ice. 628. The Vacuum System of Refrigeration. 629. The Ammonia Absorption Process. 630. The Adsorption System of Refrigeration. 631. Applications of Refrigeration. 632. Household Refrigerators.	
APPENDIX.....	1145
PROBLEMS.....	1169
INDEX.....	1187

ERRATA

HEAT-POWER ENGINEERING, PART III

BY BARNARD, ELLENWOOD, AND HIRSHFELD

PAGE

804. Line 14. Change "lb. per cuft." to cuft. per lb."
811. Line 5. Change $\frac{"(10)^5"}{3.55}$ to $\frac{"3.55"}{(10)^5}$.
844. Line 6. Change "6.03588" to "0.03588".
901. Line 12. Insert "€" before "=".
902. Line 1. After "that" add, "if $c_w = 1$."
942. Line 8 from the bottom. Change "from" to "in".
1089. Line 1. Change "is" to "are".
1097. Line 9. Change "*abedfga*" to *abedfea*."
1111. Line 9. Change "*d*" to "*p*".
1169. Prob. 17. Answer (c) should be $16.85 (10)^5$.
1171. Prob. 40. In ans. (b) add "lb." and change ans. (c) to "3.93".
1176. Prob. 16. Change ans. to "58,200" and "26,600,000."
1176. Prob. 22. Change "pressure" to "positive".
1183. Prob. 6. Insert "ideal" before "air" in the first line.
1186. Prob. 36. Change "95" to "35".

CHAPTER XLII

THE FLOW OF FLUIDS THROUGH PIPES AND PASSAGES

498. General.—(a) Engineers are constantly dealing with problems involving the flow of various fluids (such as water, steam, air, ammonia, oil, brine, and numerous other liquids, vapors, and gases) through pipe lines or other forms of passage. Sometimes this flow must take place with the fluid under considerable pressure, or the pressure may be atmospheric or less; sometimes the fluid is at a high temperature, as with gases from a furnace or steam from a superheater, or the fluid may be at a very low temperature as in refrigeration. In all cases, however, there are three items of great importance in dealing with such problems: (1) The pipe, duct, or passageway must be of **suitable strength and material** to carry the fluid without danger of rupture or serious leakage. (2) The **cross-sectional area** of the passage must be sufficient to handle the required rate of flow *economically*, i.e., there must be a correct balance between the **first cost** and **operating cost** for the known rate of flow and the number of hours per year that the apparatus is in use. (3) The **cross-sectional shape** of the passageway through which the fluid flows must be suitable, both economically and structurally, for the special purpose for which the flow is maintained.

Of the three items just enumerated, (1) and (3) involve a great variety of engineering problems that need not be considered here; but in all cases the energy used to overcome fluid friction and turbulence in the pipe line or passage must be calculated if the most economical size of line is to be obtained. Hence part (2) of the subject, assuming the steady flow of any fluid, will be treated in this chapter.

(b) The **steady flow**¹ of any fluid through any combination of apparatus between any two sections can be maintained only because of

¹ Defined in Sect. 43, Part I, as flow under the following conditions:

- (A) The weight of fluid passing any section per unit of time is constant.
- (B) The pressure, temperature, velocity, and specific volume of the fluid remain constant at any given section.
- (C) The apparatus has been operating for a sufficient length of time to have reached a stable condition, which will be maintained throughout the period in which all observations are made.

the expenditure of energy on the fluid *between* these sections; this energy may come from *one or more* of the following sources: (1) reduction in elevation, (2) drop in pressure, (3) decrease in velocity, or (4) some form of pump. The choice of the two points at which the initial and final sections are chosen in any particular case depends on the purpose of the analysis desired. Thus, for example, consider a centrifugal compressor or pump with its suction and delivery lines as indicated

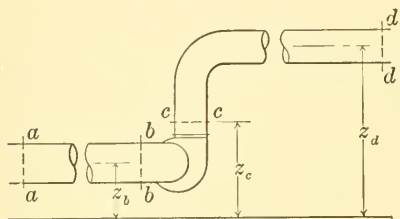


FIG. 829.

in Fig. 829. If the sections *a* and *d* are chosen some distance apart on opposite sides of the pump, the apparatus involved must then consist of the suction line, the pump or compressor, and the delivery line; and the energy equation must include all forms of energy entering and leaving the system included between the chosen

points. Instead, however, of attempting to handle the whole problem in this way, it is far simpler and much more accurate to treat the pump or compressor by itself, and each pipe line by itself.

(c) Consider the **flow of any fluid through a closed conduit**, such as a pipe line or the gas passages of a boiler, from section 1 to section 2, as indicated in Fig. 830. The total amount of energy entering the system with the fluid, during the time that *w* pounds of fluid pass section 1, may be represented by E_1 , where $E_1 = w \left(h_1 + \frac{v_1^2}{2g778} + \frac{z_1}{778} \right)$ B.t.u.,

and similarly, at the exit section during this same time the total amount of energy leaving this system *with the fluid*, in B.t.u., is

$$E_2 = w \left(h_2 + \frac{v_2^2}{2g778} + \frac{z_2}{778} \right)$$

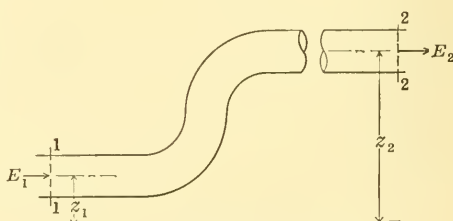


FIG. 830.

In these equations, and those that follow in this section,

g = the acceleration due to gravity = 32.17 ft. per sec.²

h = the specific heat content of the fluid, in B.t.u. per lb.

i = the specific internal energy of the fluid, in B.t.u. per lb.

v = the velocity of the fluid, in ft. per sec.

$\bar{V}$ = the specific volume of the fluid, in cu. ft. per lb.

Q_n = the net quantity of heat transferred *from* the w pounds of fluid between sections 1 and 2, due to all causes, in B.t.u.

$\rho = 1/\bar{V}$ = the density of the fluid, in lb. per cu. ft.

Since there is no pump or engine through which the fluid flows in passing from section 1 to 2, the only way by which energy may be given to, or taken from, the fluid is by heat transmission. Thus if a *steam pipe line* were involved, the term Q_n would be positive and relatively small, since it is entirely due to the uncontrolled amount of radiation, convection, and conduction of heat *from* this steam; but if the *gas passages* of a boiler were involved between sections 1 and 2, then Q_n would still be positive but very large because heat would be transferred very rapidly *from* this gas to the boiler tubes. In this case Q_n would be largely under the control of the operator who regulates the rate of firing the furnace supplying the gases under consideration. However, for any such flow, it follows from the law of the conservation of energy that

$$E_1 = E_2 + Q_n \quad . \quad . \quad . \quad . \quad . \quad (589a)$$

or

$$w \left[h_1 - h_2 + \frac{v_1^2 - v_2^2}{2g \cdot 778} + \frac{z_1 - z_2}{778} \right] = Q_n. \quad . \quad . \quad (589b)$$

Now, since $h = i + \frac{P\bar{V}}{778} = i + \frac{P}{778\rho}$, Eq. (589b) may also be expressed as

$$\frac{P_1}{\rho_1} + \frac{v_1^2}{2g} + z_1 = \frac{P_2}{\rho_2} + \frac{v_2^2}{2g} + z_2 + 778 \left[\frac{Q_n}{w} - (i_1 - i_2) \right]. \quad (590a)$$

This is an equation of **specific energy terms**, in foot-pounds per pound of fluid; and P/ρ is sometimes called the *pressure head*, $v^2/2g$ the *velocity head*, and z the *gravity head*, as in hydraulic work. However, this is a custom that may lead to confusion when the densities are appreciably different, as is often the case; and the significance of the last term requires special attention.

The term $778 \left[\frac{Q_n}{w} - (i_1 - i_2) \right]$ may be represented by the symbol x for convenience, and also because the equation may sometimes be solved for the purpose of finding x . Eq. (590a) then becomes

$$\frac{P_1}{\rho_1} + \frac{v_1^2}{2g} + z_1 = \frac{P_2}{\rho_2} + \frac{v_2^2}{2g} + z_2 + x. \quad . \quad . \quad (590b)$$

In general, x may be positive, negative, or zero. Thus, for example, if the *net amount* of heat transferred *from* the fluid between the two given

sections due to *all* causes exceeds the decrease in the internal energy of the fluid between these sections, then x will be positive. On the other hand, if heat is transferred to the fluid faster than the internal energy increases, x will be negative; and if $Q_n/w = i_1 - i_2$ then x will be zero. This discussion should serve to show clearly why x in Eq. (590b) *should not, in general, be called the loss of head due to friction*, as has very commonly been done when applying a hydraulic formula to the flow of a compressible fluid involving heat transfer.

When a *liquid* flows through a pipe line without friction, turbulence, or heat transfer, its density and internal energy remain constant; then *the sum of the pressure head, the velocity head, and the gravity head remains constant*. This is **Bernoulli's Theorem**. Note particularly the limitations imposed on the application of this theorem, and that Eqs. (589) and (590) are the more general ones dealing with *any fluid flowing through a pipe line or passage* in which turbulence, friction, or heat transfer may be involved.

One should observe that, for any fluid, the friction and turbulence in an insulated pipe line will (1) decrease the pressure, (2) decrease the available mechanical energy, (3) increase the specific volume and entropy, (4) increase the velocity, if the diameter of the line remains constant, and (5) cause the heat content and internal energy at the end of the line to be greater than they would be for a *reversible adiabatic* expansion to the same final pressure as that at the end of the line. Item (5) is especially important when dealing with the *flow of gases and vapors*, because a very small variation in the internal energy or in the heat content may easily involve a much larger amount of energy than that represented by a large change in velocity, or elevation, of a fluid. Many serious errors have thus often been introduced into the calculations for the flow of such fluids by trying to use Bernoulli's Theorem where it does not apply.

(d) As an **example** to illustrate the significance of Eq. (590), consider the flow of superheated steam through a *perfectly insulated horizontal line*, or throttling valve, from section 1 to 2.

Assume that observations show that $p_1 = 116$, and $p_2 = 25$ lb. per sq. in. abs., and that $t_1 = 340^\circ$ F. Assume the cross-section of the line at 2 to be just the right size to make $v_2 = v_1$; then $h_2 = h_1$; hence, by the aid of the steam table, the following values may be determined:

$p_1 = 116$	$p_2 = 25$
$t_1 = 340$	$t_2 = 300$
$\bar{V}_1 = 3.86$	$\bar{V}_2 = 17.84$
$h_1 = 1190.0$	$h_2 = 1190.0$
$AP_1\bar{V}_1 = 82.8$	$AP_2\bar{V}_2 = 82.5$
$i_1 = 1107.2$	$i_2 = 1107.5$

Then, from Eq. (590a), since $v_1 = v_2$, $z_1 = z_2$, and $Q_n = 0$, it follows that

$$\frac{P_1}{\rho_1} - \frac{P_2}{\rho_2} = 778 [0 - (1107.2 - 1107.5)] = 233.4 \text{ ft.-lb. per lb. of steam.}$$

Clearly, with such a change in density as occurs here ($17.84 \div 3.86 = 4.62$), to try to express this result in *feet* of the fluid immediately calls forth the question: What is the density of the fluid involved?

Next assume that the flow of steam takes place with p_1 , t_1 , and p_2 as before, but that now $Q_n/w = 9.8$ B.t.u. per lb. Then $h_2 = h_1 - 9.8 = 1180.2$, and consequently, from the steam table, $t_2 = 280$ and $\bar{V}_2 = 17.33$; hence $AP_2\bar{V}_2 = 80.2$, $i_2 = 1100$, and

$$\frac{P_1}{\rho_1} - \frac{P_2}{\rho_2} = 778 [9.8 - (1107.2 - 1100)] = 2023 \text{ ft.-lb. per lb.}$$

Here the value of x is about 9 times its previous one; although in both examples, $p_1 - p_2 = 116 - 25 = 91$ lb. per sq. in.; and since $v_1 = v_2$, and $z_1 = z_2$ in both cases, these examples show that *the term x is not in general a friction head*.

(e) The flow of hot gases over the heating surfaces of a steam-generating unit is an important engineering problem; Eqs. (589) and (590) will hold for such a case, but their use is generally of little value in trying to find the *friction loss* of these gases. The reason for this will now be explained. The amount of heat absorbed from the hot gases in passing over the heating surfaces is likely to range from 250 to 400 B.t.u. per lb., but the energy required to produce the flow of gas past these surfaces will generally be only from $\frac{1}{10}$ to $\frac{1}{2}$ of 1 per cent as much; hence the energy involved in the latter is far less than the errors involved in the measurement of the large energy terms of these equations. On the other hand, if the flow of various kinds of fluid through pipes and ducts be studied carefully under such conditions that the amount of heat transferred to or from the fluid is negligibly small, then the amount of energy expended to overcome fluid friction may be accurately measured. Such information may thereafter be *used to estimate roughly the fluid friction* losses that must be overcome in forcing gases, vapors, and liquids over various kinds of heat-transfer surfaces, or through various arrangements of pipes and ducts. The problem becomes very complex when fluid friction and heat transfer are both involved, as may be seen from the following section.

499. Resistance to the Flow of Fluids in "Short" and "Long" Lines.—(a) The terms *short* and *long* may very naturally be used to indicate the relative lengths of pipe lines; but they are also used in a broader sense when dealing with the flow of fluids through passageways and pipes. Then they represent the two classes into which flow problems are divided, the distinction depending upon whether a simple

“short line” formula will be sufficiently accurate, or whether a more complicated one should be used. The actual length of the line, however, is not the real factor which always determines this classification, as the succeeding treatment will show.

(b) Consider **any fluid** flowing under steady flow conditions in a *horizontal* pipe or passage of inside diameter D , across an elementary section of length dL , thereby causing a drop in pressure, which may be expressed as an increment of pressure, $-dP$. Experiments have shown that, for short distances with all kinds of flow without appreciable change in the fluid density, the frictional resistance is closely proportional to the density (ρ) of the fluid, the velocity head ($v^2/2g$), and the area of the surface (πDdL) in contact with the fluid. Therefore, this resistance may be expressed as

$$dF = c\rho \frac{v^2}{2g} (\pi DdL), \quad . \quad . \quad . \quad . \quad . \quad . \quad (a)$$

where c is a coefficient of resistance that is constant for any given density, viscosity, velocity, diameter, and pipe surface, and is to be found from experimental data given later. Since this force dF acts on the fluid during its movement through the length dL , the energy expended in *overcoming this resistance* is

$$dF \cdot dL = \left(c\rho \frac{v^2}{2g} \pi DdL \right) dL. \quad . \quad . \quad . \quad . \quad . \quad . \quad (b)$$

This amount of energy is converted into heat which will be entirely absorbed by the fluid if the pipe line is perfectly insulated; but if the pipe is not insulated a large part of this energy may be dissipated to the medium surrounding the pipe line, and the rest will be retained by the fluid, the amount depending on the temperature of the fluid relative to that of the surrounding medium.

Since, in general, any fluid is compressible, its specific volume will increase when its pressure is lowered, assuming its temperature to remain constant. If the temperature rises because of heat transferred to the fluid from an external source, the increase in specific volume will be still more than that due to the drop in pressure. In one special case the specific volume could remain constant; viz., if heat is transferred *from* the fluid at just the right rate to offset the tendency of the volume to increase due to the drop in pressure; but this will seldom occur, so the specific volume may, in general, be assumed to increase. Therefore, *the velocity will increase* from v to $v + dv$, assuming the pipe to be of uniform diameter. This *acceleration of the fluid*, due to the increase in

its specific volume, involves an absorption of kinetic energy by the fluid equal ² to

$$(\text{Weight of fluid}) \left[\frac{(v + dv)^2 - v^2}{2g} \right] = \frac{\rho \pi D^2 dL}{4} \left[\frac{v dv}{g} \right]. \quad (c)$$

The sum of the energies given by Eqs. (b) and (c) represents the amount of *net work done* on this small section of fluid in forcing it across the elementary section due to the increment in pressure, $-dP$. But

this *net work* is also equal to $(-dP) \left(\frac{\pi D^2}{4} \right) dL$.

Therefore,

$$(-dP) \left(\frac{\pi D^2}{4} \right) dL = \left(c \rho \frac{v^2}{2g} \pi D dL \right) dL + \left(\frac{\rho \pi D^2}{4} dL \right) \left(\frac{v dv}{g} \right). \quad (d)$$

or

$$-dP = 4c \frac{\rho}{D} \frac{v^2}{2g} dL + \frac{\rho v dv}{g}. \quad (e)$$

Then integrating between sections 1 and 2, Eq. (e) becomes

$$\int_{P_1}^{P_2} -dP = \int_{L_1}^{L_2} \frac{4c\rho}{D} \frac{v^2}{2g} dL + \int_{v_1}^{v_2} \frac{\rho v dv}{g}. \quad (f)$$

(c) For “**short**” horizontal lines, in which the length ($L_2 - L_1$) is sufficiently short, and the heat transferred to or from the fluid is very small, so that the density, velocity, and the coefficient c may be treated as constants in Eq. (f), by integration,

$$P_1 - P_2 = 4c \frac{\rho}{D} \frac{v^2}{2g} (L_2 - L_1). \quad (g)$$

If L be used to represent this length then Eq. (g) becomes

$$P_1 - P_2 = 4c \frac{L}{D} \rho \frac{v^2}{2g}. \quad (591a)$$

This is the form of an equation that is of great use in calculating the friction drop in pressure in all so-called “**short lines**” which, in general, may be considered as those conduits, pipes, or passages of uniform diameter, through which the fluid flows without changing its density more than about 10 per cent from any cause. Note (1) that for ordinary water lines of constant diameter this formula therefore applies for all lengths; (2) that for gas lines in which the temperature and diameter are approximately constant the formula will be accurate only when the drop in pressure is less than about 10 per cent of the initial absolute pressure; and

² Since $(dv)^2 \cdot dL$ is a negligible quantity.

(3) *that for liquids, gases, or vapors flowing in pipes or tubes of uniform diameter and subject to rapid rates of heat transfer, the density may thereby be changed more in a few feet of line than it would be by friction in several miles of ordinary pipe kept at constant temperature.* However, if the density be changed not more than about 30 per cent, the formula may still be applied with a fair degree of accuracy by using the *average density* and the *average velocity* in the line.

In all *pipe lines in steam power plants* the drop in pressure is seldom greater than 10 per cent of P_1 , and the rate of heat transfer is very low when the line is well insulated.

If a pipe line has a *difference in elevation* between the two points for which the drop in pressure is calculated, the proper allowance must then be made for this difference.

(d) For "**long lines**," the evaluation of Eq. (f), page 787, cannot be easily made, in general, because, c , ρ , and v may vary appreciably. However, for the case of the *flow of a gas or vapor* that follows the law ³ $PV = wRT$, the integration of Eq. (e) may be made for one case as follows:

The velocity at a section having the area a is

$$v = \frac{\text{total volume}}{\text{area}} = \frac{V}{a} = \frac{wRT}{aP}; \text{ also } \frac{1}{\rho} = \bar{V} = \frac{RT}{P}.$$

Hence,

$$\frac{v^2}{2g} = \frac{w^2 R^2 T^2}{2ga^2 P^2}.$$

Then, by substituting these values and transposing, Eq. (e) becomes

$$- \left(\frac{2ga^2 P^2}{w^2 R^2 T^2} \right) \left(\frac{RT}{P} \right) dP = \frac{4c}{D} dL + \left(\frac{2g}{v^2} \right) \frac{v dv}{g}$$

or

$$- \left(\frac{2ga^2}{w^2 R T} \right) P dP = \frac{4c}{D} dL + \frac{2dv}{v}. \quad . \quad . \quad . \quad . \quad (h)$$

Now, if the *temperature of the fluid be constant*, and if the *coefficient c be constant*, the integration of Eq. (h) between sections 1 and 2, the intervening length being L , gives

$$\frac{2ga^2}{w^2 R T} \left[\frac{P_1^2 - P_2^2}{2} \right] = \frac{4cL}{D} + 2 \log_e \left(\frac{v_2}{v_1} \right)$$

or

$$P_1^2 - P_2^2 = \frac{w^2 R T}{ga^2} \left[4c \left(\frac{L}{D} \right) + 2 \log_e \frac{v_2}{v_1} \right]. \quad . \quad . \quad (591b)$$

³ For a mixture of gases, see Eq. (533), page 406.

For long-distance transmission of air and other gases this form of the equation is fairly satisfactory because the temperature is nearly constant even though the density varies enormously, and an *average* value of c , for any given pipe diameter and surface, may be used.⁴

(e) If heat is transferred rapidly to or from a fluid flowing in a passageway, the drop in pressure of this fluid *cannot, in general, be accurately found* from either Eq. (591a) or (b) because the temperature is likely to be far from constant. However, these equations are sometimes applied as *approximations*, by using the best available average values of ρ , v , and c for the given case. These approximations are well justified in such cases as those of the gas passages from a furnace, because the variable accumulation of scale and ash affects the temperature, velocity, and coefficient very appreciably, and consequently an approximation is all that is needed.

(f) The values of the coefficient c must be known before Eq. (591 a) or (b) can be used. Experiments show that these values depend upon the *viscosity*, *density*, and *velocity* of the fluid, also upon the *diameter* of the pipe and the *roughness* of its surface. Fortunately this coefficient can be found in terms of these characteristics of fluid flow that are perfectly general, so that the coefficient thus determined will hold for *any fluid*. Since viscosity and its units are of the utmost importance in dealing with the flow of fluids, these points will be considered before the general method of finding the coefficient is taken up.

500. Viscosity and Its Units.—(a) The **absolute viscosity** of a fluid, or the **coefficient of viscosity**, is defined as the tangential force required to move a plane surface of the fluid, of unit area, with unit velocity relative to another parallel plane surface of the fluid at unit distance from the first. In the c.g.s. system the unit of viscosity is the *poise*, which is 1 dyne sec. per sq. cm. One *centipoise* = 0.01 poise. No name for the unit of absolute viscosity in the English system has been generally agreed upon, and thus the c.g.s. unit is commonly used. The usual symbol for absolute viscosity is the Greek letter mu (μ).

(b) The **kinematic viscosity**⁵ of a fluid is defined as the absolute viscosity divided by the density of the fluid, and is expressed in square centimeters per second or square feet per second. The symbol generally used for kinematic viscosity is the Greek letter nu (ν) or the English letter γ . The latter is considered preferable by the authors and will

⁴ In Eqs. 591 (a) and (b) *consistent units* would be: P in lb. per sq. ft., w in lb. per sec., R in ft.-lb. per lb. per °F., T in deg. F., abs., g in ft. per sec.², a in sq. ft., D and L in ft., v in ft. per sec., and ρ in lb. per cu. ft.

⁵ Kinematic viscosity is sometimes also defined as the ratio of absolute viscosity to the specific gravity, but this definition is not recommended by the A.S.M.E.

therefore be used in this text. If the values of the kinematic viscosity of a fluid for various temperatures and pressures are available in convenient form, they are very useful in making pipe line calculations. For certain fluids the kinematic viscosities are given in the form of curves in Figs. 834, 835 and 836, pages 797-799.

(c) The **relative viscosity** of any fluid is the ratio of the absolute viscosity of this fluid to that of water. Its symbol, as given by the A.S.M.E., is z .

(d) The **density** of a substance is defined as *mass* per unit volume; and it is also defined as *weight* per unit volume.⁶ Its symbol is the Greek letter rho (ρ). The use of the word "weight" in this definition is clearly to designate the amount of matter per unit volume; or in other words, for this case the "weight" of a known volume of the fluid is to be found by balancing against standard "weights" or lumps of metal. (In another and stricter sense, the "weight" of a body is the force of gravity on that body in any given locality, as determined by the elongation of a spring supporting that body at the given place.)

(e) The **pound mass** and the **pound force** are often confused when dealing with viscosity units. The *pound mass* is defined as the mass of a certain lump of metal kept as the standard. The *pound force* is defined as the force of gravity on the pound mass in the standard locality, i.e., where the acceleration due to gravity is 32.174 ft. per sec.² The pound force may also be equally well defined as that force which will give an acceleration of 32.174 ft. per sec.² to the pound mass. Consequently, if the pound force and the pound mass are both to be used in the equation of motion, the factor 32.174 is directly involved in this equation. Thus, 1 lb. force = (1 lb. mass) (32.174 ft. per sec.²)
= (32.174 lb. mass)(1 ft. per sec.²).

Hence for the pound force to produce the unit acceleration on a *unit* of mass, the latter needs to be 32.174 times as big as the pound mass. Such a unit of mass is called the **slug**, and the mass of a body in *slugs* is equal to its mass in pounds $\div$ 32.174.

(f) An **absolute system of units** is one for which the fundamental equation between the units of force and mass is:

$$\text{A unit force} = (\text{a unit mass})(\text{a unit acceleration}).$$

If the unit force be the *dyne*, the unit mass becomes the *gram* and the unit acceleration 1 cm. per sec.²

If the unit force be the *pound*, then this equation requires that the unit mass shall be the *slug*, and the unit acceleration 1 ft. per sec.²

⁶ See A.S.M.E. Test Code on Definitions and Values.

If the unit mass be the *pound*, the above equation requires that the unit force shall be the $1/32.174$ th part of the pound force, or the *poundal*.

(g) The **conversion factors** which follow are needed when dealing with viscosity in various systems of units.

$$\begin{aligned}
 1 \text{ cm.} &= 0.03281 \text{ ft.} & 1 \text{ lb. force} &= 32.174 \text{ poundals} \\
 1 \text{ cm.}^2 &= 1.076 (10)^{-3} \text{ sq. ft.} & &= (1 \text{ lb. mass}) (32.174 \text{ ft. per sec.}^2) \\
 1 \text{ cm.}^3 &= 3.53 (10)^{-5} \text{ cu. ft.} & 1 \text{ lb. mass} &= \frac{1 \text{ lb. force}}{32.174 \text{ ft. per sec.}^2} \\
 1 \text{ gram} &= 220.5 (10)^{-5} \text{ lb. mass.} & & \\
 1 \text{ dyne} &= \frac{1}{980665} \text{ kg. force} & 1 \text{ slug} &= 32.174 \text{ lb. mass.} \\
 &= 2.248 (10)^{-6} \text{ lb. force.} & & \\
 1 \text{ poise} &= 100 \text{ centipoises.} & & \\
 &= \frac{1 \text{ dyne sec.}}{\text{sq. cm.}} = \left(\frac{1}{98.07} \right) \frac{\text{kg. force sec.}}{\text{sq. m.}} = \left(\frac{1}{10} \right) \frac{\text{kg. mass}}{\text{m. sec.}} \\
 &= 0.0672 \frac{\text{lb. mass}}{\text{ft. sec.}} \quad \text{or} \quad \left(\frac{1}{14.88} \right) \frac{\text{lb. mass}}{\text{ft. sec.}} \\
 &= 0.0672 \frac{\text{poundal sec.}}{\text{sq. ft.}} \quad \text{or} \quad \left(\frac{1}{14.88} \right) \frac{\text{poundal sec.}}{\text{sq. ft.}} \\
 &= 0.00209 \frac{\text{lb. force sec.}}{\text{sq. ft.}} \quad \text{or} \quad \left(\frac{1}{479} \right) \frac{\text{lb. force sec.}}{\text{sq. ft.}}.
 \end{aligned}$$

To the engineer who commonly deals with “the pound” without thinking whether he is talking about the pound force or the pound mass, these conversions may at first appear complicated; but the distinction between the two uses of this word becomes necessary when dealing with viscosities. Another advantage of having the complete set of viscosity units is to enable one to check published values, which sometimes are given in wrong units and in such an incomplete manner that one needs all these various forms to ascertain what is meant. The viscosity unit is not a simple one with any system of fundamental units; but the importance of the term in the flow of fluids is gradually becoming appreciated by engineers.

Some examples of the use of the preceding conversion factors as needed in the flow of fluids are given in Table LXVIII, page 796 (also in Fig. 832). The viscosities as found from viscosity tables are generally given in poises or centipoises, and these values may sometimes be used in engineering work without conversion to other units. The conversion factors as here given will, however, enable the English units to be found readily when needed.

(h) The effect of temperature on the absolute viscosity of most fluids is very noticeable. Most liquids have their absolute viscosities materially decreased by an increase in temperature, as shown by the curves in Fig. 831. On the other hand, an increase in the temperature of gases and vapors causes their absolute viscosities to increase, as shown by the curves in Fig. 832.

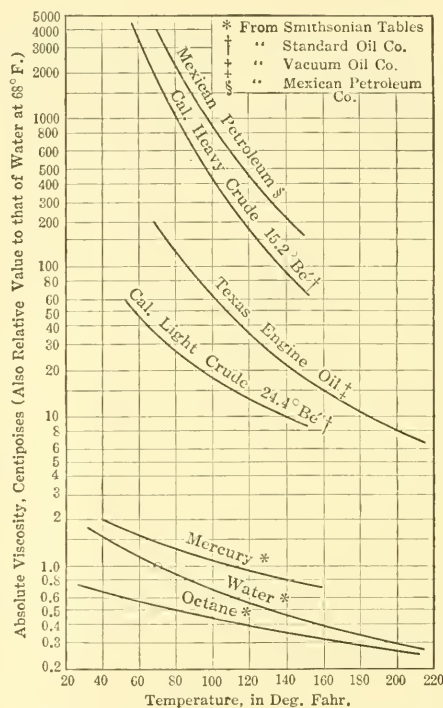


FIG. 831.—Absolute Viscosities of Liquids.

The general relation between the absolute viscosity (μ) of a gas or vapor at any absolute temperature (T in deg. cent.) and its known viscosity (μ_o) at some absolute temperature (T_o in deg. cent.) is not a simple one. However, the *Sutherland formula* is known to give very satisfactory results for this relation; this formula is

$$\mu = \mu_o \left(\frac{T_o + C}{T + C} \right) \left(\frac{T}{T_o} \right)^{1.5} \quad \dots \quad (592)$$

In this equation all the symbols have just been defined except C , which is a constant for any given fluid and must be determined by experiment.

The values of this constant, for some important **gases and vapors**, as given by the International Critical Tables and by Landolt and Börnstein Tables (1931), are as follows:

Fluid	Air	CO ₂	CO	CH ₄	H ₂	O ₂	N ₂	NH ₃	SO ₂
<i>C</i>	120	240	118	198	72	127	110.6	370	416

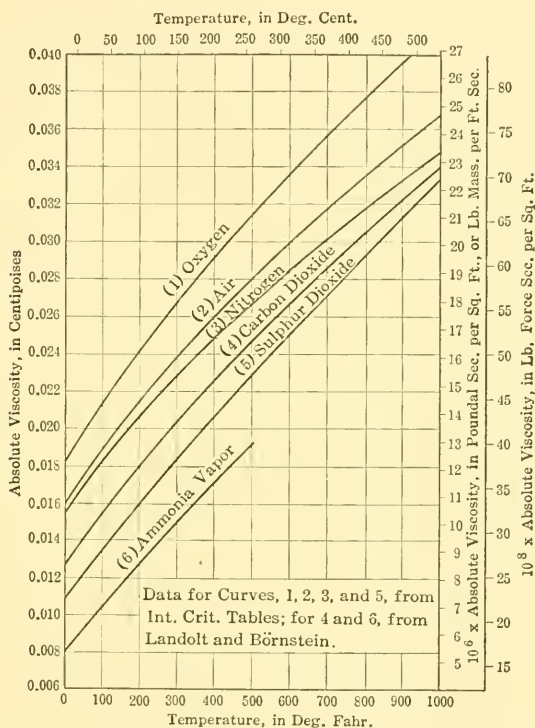


FIG. 832.—Absolute Viscosities of Gases and Vapors at Atmospheric Pressure.

If the absolute temperatures in the Sutherland formula, Eq. (592), are expressed in degrees Fahrenheit, the values of *C* given in the above table should be multiplied by 1.8 for use in the equation.

(i) The **effect of pressure on the absolute viscosity of gases** is generally too small to be of much importance in engineering work. However, when working with extremely high pressures this effect should be considered. For example, Landolt and Börnstein give a value of the absolute viscosity of nitrogen under a pressure of 176.9 atmospheres and a temperature of 30 deg. cent. (86 deg. fahr.) as 0.026 centipoise, which is approximately 45 per cent greater than its viscosity

at the same temperature and under atmospheric pressure, as shown in Fig. 832. For a pressure of 73 atmospheres the same reference shows that the viscosity of nitrogen at 30 deg. cent. is 11.4 per cent greater than that at atmospheric pressure.

For **steam** the absolute viscosity is more dependent on pressure than is generally realized. The data on this point are rather meager, but recently the experimental results of Dr. Ing. H. Speyerer of Vienna have become available.⁷ These results are shown in Fig. 833. From these curves the absolute viscosity of *superheated* steam under a pressure of 10 atmospheres is seen to be from 8 to 13 per cent greater than that of superheated steam of the same temperature at 1 atmosphere; the higher the temperature the less is the percentage difference. These

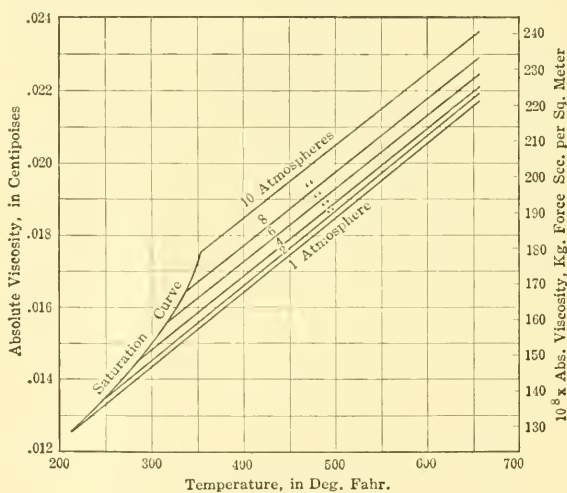


Fig. 833.—Absolute Viscosity of Steam. (Data from Speyerer.)

curves also show that *saturated steam* at 10 atmospheres has an absolute viscosity 40 per cent greater than that of saturated steam at 1 atmosphere; but of course this large difference is primarily due to the greater saturation temperature at the higher pressure. It should also be noted that any given percentage increase of pressure causes a much greater increase in the viscosity the higher the pressure. For example, keeping the temperature constant but increasing the pressure from 2 atmospheres to twice that value will cause the viscosity to be only about 2 per cent greater; but changing the pressure from 5 atmospheres to twice that value will increase the viscosity about 7.5 per cent.

⁷ See Z.V.D.I., 1925, Vol. 69, p. 747.

501. Reynolds' Number.—(a) The work of Professor Osborne Reynolds regarding the *critical velocity* of a fluid in a tube has already been mentioned (p. 260); this velocity is defined as that at which the flow changes from a *viscous* to a turbulent one, and was shown by Reynolds, in 1874, to vary directly as the absolute viscosity of the fluid and inversely as the tube diameter and the density of the fluid. These three factors combined with the average velocity of the fluid in a pipe are now known to be the chief ones in determining the friction factor that is to be used when calculating the resistance to the flow of any fluid in any kind of a conduit or pipe line.

The work of Reynolds has been supplemented by many others⁸ regarding the flow of fluids in pipes; but Lord Rayleigh⁹ in 1892 was the first one to apply the principle of *dynamical similarity* to flow of fluids in pipe lines, or to the flow of a fluid past any body immersed in it, and thus derive the general relation regarding the friction factor and a certain ratio called **Reynolds' number**. Lord Rayleigh was careful to mention that this principle requires that the surfaces relative to which the fluid flows must not only be *geometrically similar*, but they must also have the *same degree of roughness*. He showed that, in general, when this is true the resistance to the flow of a fluid past a body depends only on the linear dimension of the body, the velocity, density, and viscosity of the fluid. The importance of these four terms cannot be over-estimated in studying the flow of fluids in pipes, and a definite combination of them is now commonly expressed as *Reynolds' number*, or the *turbulence factor*, as follows:

$$\text{Reynolds' number} = \frac{(\text{diameter})(\text{velocity})(\text{density})}{\text{absolute viscosity}} = \frac{Dvp}{\mu} \quad (593)$$

$$= \frac{(\text{diameter})(\text{velocity})}{\text{kinematic viscosity}} = \frac{Dv}{\nu} \quad . \quad . \quad . \quad (594)$$

Reynolds' number is a *dimensionless ratio*, and this adds very materially to the convenience of its general application in all countries, as will now be shown; and then the method of finding the friction factor by means of this number will be considered.

(b) **Any homogeneous system of units** will give the same value of Reynolds' number, because it is dimensionless. An example to illus-

⁸ See "Similarity of Motion in Relation to the Surface Friction of Fluids," by T. E. Stanton and J. R. Pannell, Phil. Trans. of the Royal Society, Vol. 214, 1914, p. 199.

⁹ Phil. Magazine, Vol. 34, 1892, p. 59.

trate this is given in Table LXVIII. The conversion factors, as given on page 791, are needed in such examples.

TABLE LXVIII
AN EXAMPLE OF THE CALCULATION OF REYNOLDS' NUMBER

Quantity	Symbol	System of Units		
		Absolute Metric Cm., Gram, Sec. "C.G.S." System	Absolute English Foot, Pound Mass, Sec.	Absolute English Foot, Pound Force, Sec.
Diameter	D	2.5 cm.	0.0821 ft.	0.0821 ft.
Velocity	v	600 cm./sec.	19.7 ft./sec.	19.7 ft./sec.
Density	ρ	0.9 gram/cm. ³	56.2 $\frac{\text{lb. mass}}{\text{cu. ft.}}$	$\frac{56.2}{32.17}$ or 1.748 $\frac{\text{slugs}}{\text{cu. ft.}}$
Absolute Viscosity	μ	0.25 poise or $0.25 \frac{\text{dyne sec.}}{\text{cm.}^2}$	0.0168 $\frac{\text{lb. mass}}{\text{ft. sec.}}$ or $0.0168 \frac{\text{poundal sec.}}{\text{sq. ft.}}$	0.000522 $\frac{\text{lb. force sec.}}{\text{sq. ft.}}$
Kinematic Viscosity = μ/ρ	ν or y	0.278 cm. ² /sec.	0.000299 $\frac{\text{sq. ft.}}{\text{sec.}}$	0.000299 $\frac{\text{sq. ft.}}{\text{sec.}}$
Reynolds' Number = $Dv\rho/\mu$ = Dv/y	R_n	5400	5400	5400

(c) A hybrid system of units often has many advantages when finding Reynolds' number with the usual units used by engineers in this country, because the *centipoise* is the c.g.s. unit of absolute viscosity that is commonly available in standard tables. If the absolute English system (MLT) be used for all terms except viscosity, which is in centipoises, there results the following equation:

Reynolds' number = $\frac{1488(D \text{ in ft.})(v \text{ in ft./sec.})(\rho \text{ in lb./cu. ft.})}{\mu \text{ in centipoises}}$. (595)

Thus, for the example in Table LXVIII, Eq. (595) yields

$$\text{Reynolds' number} = \frac{1488(0.0821)(19.7)(56.2)}{25} = 5400,$$

which checks the previous results.

Since Eq. (595) already has a conversion factor incorporated in it, one may also include another factor and express the *pipe diameter in inches* by d , thus giving the following:

$$\text{Reynolds' number} = \frac{124(d \text{ in in.})(v \text{ in ft./sec.})(\rho \text{ in lb./cu. ft.})}{\mu \text{ in centipoises}}. \quad (596)$$

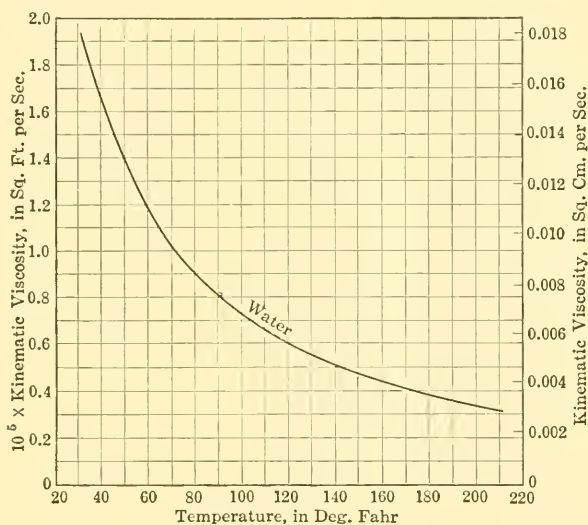


FIG. 834.—Kinematic Viscosity of Water.

Eq. (596) is probably the most convenient form for general engineering use in this country when dealing with any fluid, provided its kinematic viscosity is not available for the given pressure and temperature.

(d) The use of the kinematic viscosity (y), when available, will shorten the calculation of Reynolds' number very materially, as indicated by Eq. (594). Since the flow of water, steam, air, nitrogen, and carbon dioxide is so frequently encountered in power plant work, the values of their kinematic viscosities have been calculated by the authors

and the results are given by the curves in Figs. 834, 835, and 836, for a considerable range of temperatures. These values are all for a pressure of 1 atmosphere except those for air and steam, which have the various pressures as shown on the curves.

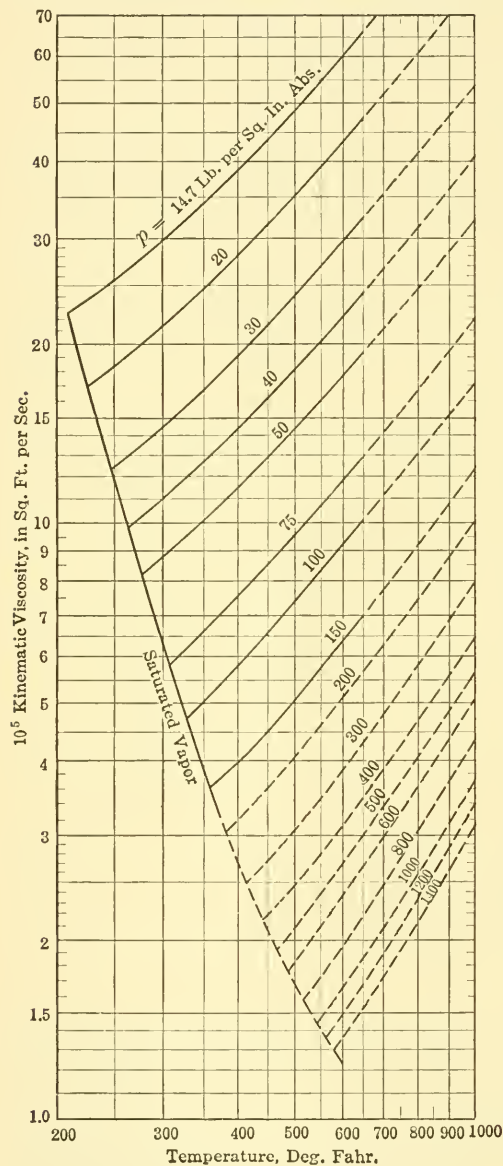


FIG. 835.—Kinematic Viscosity of Steam.

Special attention is called to the lower part of Fig. 835, which shows the **probable values of the kinematic viscosity of steam at high pressures**, as estimated by the authors of this text. The broken lines indicate that the values are beyond the range of experimental data thus far available. In making this extrapolation the saturation curve was extended as a smooth curve, and then, the saturation temperature being known, one point was thus determined for each pressure curve. The slope of each high pressure curve, however, had to be estimated from the other curves.

In determining the kinematic viscosity directly from the Saybolt or Engler viscosimeters, for which the time of flow of a given fluid is t seconds, the following equations¹⁰ may be used:

¹⁰ For Eqs. (597a) and (598a) see "Standardization of Saybolt Universal Viscosimeter," by Herschell, U. S. Bureau of Standards Scientific Paper 112.

For the Saybolt, the kinematic viscosity is

$$y = \frac{\mu}{\rho} = 0.0022t - \frac{1.8}{t}, \quad \frac{\text{sq. cm.}}{\text{sec.}} \quad . \quad . \quad . \quad (597a)$$

$$= \left[0.236t - \frac{194}{t} \right] 10^{-5}, \quad \frac{\text{sq. ft.}}{\text{sec.}} \quad . \quad . \quad (597b)$$

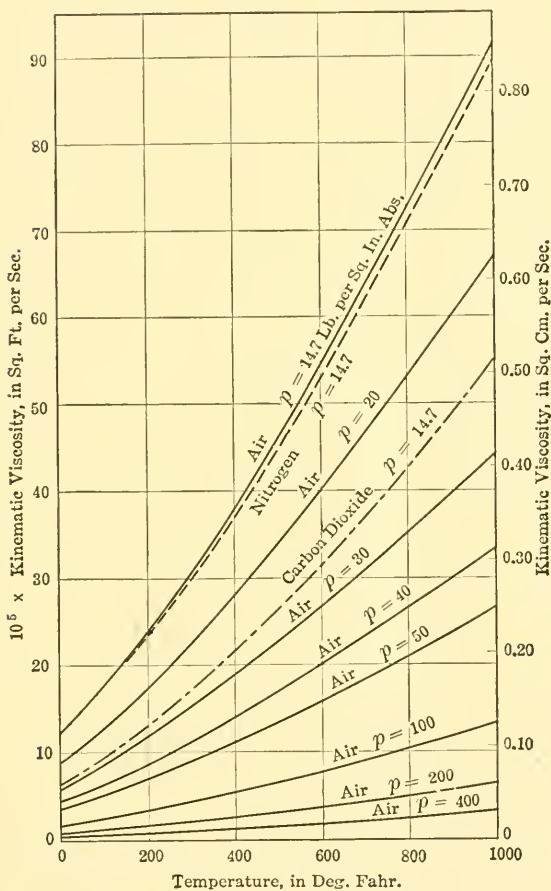


FIG. S36.—Kinematic Viscosity of Gases.

For the Engler,

$$y = \frac{\mu}{\rho} = 0.00147t - \frac{3.74}{t}, \quad \frac{\text{sq. cm.}}{\text{sec.}} \quad . \quad . \quad . \quad (598a)$$

$$= \left[0.158t - \frac{402}{t} \right] 10^{-5}, \quad \frac{\text{sq. ft.}}{\text{sec.}} \quad . \quad . \quad (598b)$$

Since the Saybolt instrument is commonly used in this country, Eq. (597b) has been plotted as shown by the curve in Fig. 837.

The use of the kinematic viscosity curves that are given in this section, and others that may be drawn when needed, will enable Reynolds' number to be found quickly, and then this number may be used to advantage to estimate the friction loss in a pipe line, as explained in the succeeding sections.

502. Viscous Flow and the Friction Factor.—(a) In viscous or stream-line flow, each particle of the fluid moves in a direction parallel to the motion of every other particle. The magnitude of the velocity of

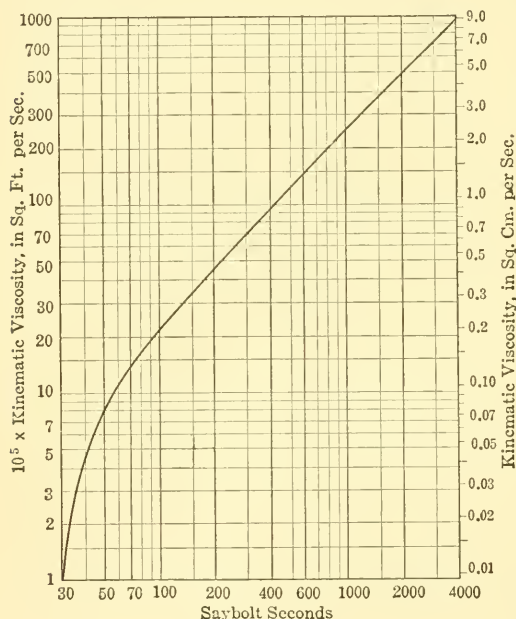


FIG. 837.

the particle increases with the distance of the particle from the pipe surface; and the average velocity over the entire cross-section of a circular pipe is equal to one-half the maximum value which is at the center of the pipe. This fact together with the equation expressing *Poiseuille's law* of viscous flow may be derived, directly, from an application of the definition of absolute viscosity.

Consider a short horizontal pipe of inside diameter D and length L , within which a cylinder of fluid with radius r is

moving relative to the adjacent fluid layer. Let the radial velocity gradient at the radius r be dv_r/dr , and let the absolute viscosity be assumed constant with a value μ . The pressure drop $(P_1 - P_2)$ along the pipe produces a force on the cylinder of fluid in the direction of motion equal to $(P_1 - P_2)\pi r^2$, as indicated in Fig. 838; also the viscous shearing forces acting on the cylinder of fluid due to the adjacent layers are equal to the product of the absolute viscosity, the shearing area, and the radial velocity gradient. Now if there be *no acceleration of the fluid in any layer*, the force due to the drop in pressure must be equal and opposite to that due to shearing,

or

$$(P_1 - P_2)\pi r^2 = -2\pi r L \mu \frac{dv_r}{dr}. \quad . \quad . \quad . \quad (599)$$

Hence,

$$dv_r = -\frac{(P_1 - P_2)}{2L\mu} r dr.$$

Assuming that there is no slip of the fluid at the inside surface of the pipe, i.e., $v_r = 0$ when $r = D/2$, the velocity v_r of the fluid at the radius r may then be found by integration. Thus, for any given value of $P_1 - P_2$, L , and μ ,

$$\int_0^{v_r} dv_r = -\frac{(P_1 - P_2)}{2L\mu} \int_{\frac{D}{2}}^r r dr$$

$$v_r = \frac{(P_1 - P_2)}{4L\mu} \left(\frac{D^2}{4} - r^2 \right). \quad . \quad . \quad . \quad (600)$$

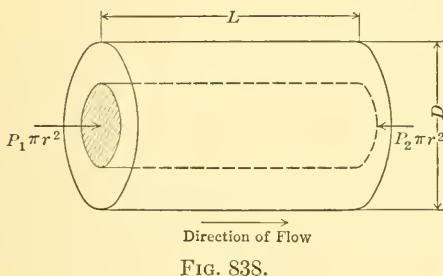


FIG. 838.

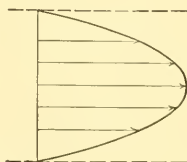


FIG. 839.—Velocity Distribution in Viscous Flow.

From this equation the corresponding *velocity distribution across the pipe* may be plotted for viscous flow, as shown by the curve in Fig. 839. This curve is a parabola with the vertex at the axis of the pipe.

For the flow of a fluid in a pipe line, the **average velocity**, v , is defined as the volume rate of flow, V , divided by the cross-sectional area of the pipe, A .

The **volume rate of flow** of the fluid may be found as follows:

$$V = \int_0^{\frac{D}{2}} v_r \cdot 2\pi r dr = \frac{(P_1 - P_2)}{4L\mu} \cdot 2\pi \int_0^{\frac{D}{2}} \left(\frac{D^2 r}{4} - r^3 \right) dr$$

$$= 2\pi \frac{(P_1 - P_2)}{4L\mu} \cdot \frac{D^4}{64}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (601)$$

Thus

$$\frac{\Delta P}{\rho v^2} = \frac{32\mu}{Dv\rho} \cdot \frac{L}{D} = \frac{32y}{Dv} \cdot \frac{L}{D} \quad \cdot \cdot \cdot \cdot \cdot \quad (605a)$$

Then

$$\frac{\Delta P}{\rho v^2} \cdot \frac{D}{L} = \frac{32\mu}{Dv\rho} = \frac{32y}{Dv} \quad \cdot \cdot \cdot \cdot \cdot \quad (605b)$$

Poiseuille's law is often written in terms of a *dimensionless variable*, the **friction factor**, f . This factor is defined in several ways, but for this text, as in many other cases, the *friction factor is obtained for either viscous or turbulent flow in short pipes or circular conduits by solving for f in the following equation for the pressure drop due to friction:*

$$\Delta P = f \left(\frac{L}{D} \right) \frac{\rho v^2}{2} \quad \cdot \cdot \cdot \cdot \cdot \quad (606)$$

Then, from Eqs. (605b) and (606), it follows that for *viscous flow in circular conduits and pipes*, the friction factor becomes

$$f = \frac{64y}{Dv} = \frac{64}{\text{Reynolds' number}} \quad \cdot \cdot \cdot \cdot \cdot \quad (607)$$

The friction factor may be plotted as ordinate versus Reynolds' number (Dv/y) as abscissa, and the hyperbolic curve shown in Fig. 840 results when viscous flow is involved.

To use Fig. 840 to determine the pressure drop accompanying *viscous flow* in any given case, Reynolds' number should first be calculated, then the ordinate, f , read. Next the pressure drop may be found from Eq. (606). *In this equation any homogeneous system of units may be used.*

In this country the absolute F.L.T. English system will be found convenient. With this system,

ΔP = pressure drop in the pipe line, in *lb. per sq. ft.*
 f = friction factor (dimensionless).

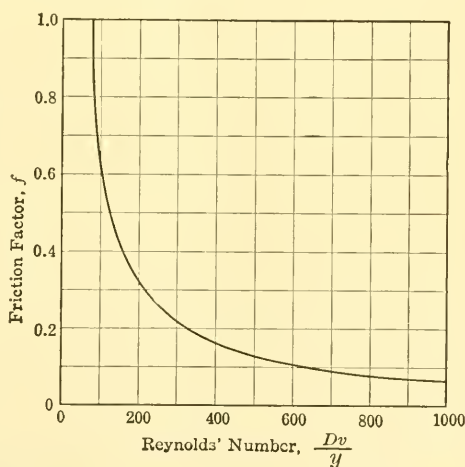


FIG. 840.—The Variation of f with Reynolds' Number for Viscous Flow.

L = equivalent length of straight pipe, in ft. (See Sect. 504 p. 812 for the effect of bends and fittings.)

D = inside diameter of pipe, in ft.

ρ = density of fluid, in slugs per cu. ft.

v = average velocity of flow, in ft. per sec.

If the density (ρ) is expressed in *pounds per cubic foot*, as is often the case, the term (g) should appear in the denominator of Eq. (606), which then becomes ¹¹

$$\Delta P = f \left(\frac{L}{D} \right) \frac{\rho v^2}{2g}. \quad . \quad . \quad . \quad . \quad . \quad (608)$$

The *following additional units* are also convenient in engineering work:

Δp = drop in pressure, in *lb. per sq. in.*

d = inside of diameter of the pipe, in in.

$\bar{V}$ = $1/\rho$ average specific volume of the fluid, in lb. per cu. ft.

g = 32.17 ft. per sec².

Using these units, Eq. (608) becomes

$$\Delta p = \frac{fLv^2}{772d\bar{V}} = \frac{0.001295fLv^2}{d\bar{V}}. \quad . \quad . \quad . \quad . \quad (609)$$

Very often the rate of flow, w , in *lb. per sec.*, is more convenient to use than the velocity, v ; then Eq. (609) becomes

$$\Delta p = \frac{43.7fL\bar{V}w^2}{d^5}. \quad . \quad . \quad . \quad . \quad . \quad (610a)$$

For large conduits, this equation is more convenient when the diameter is kept in feet, D . Then

$$\Delta p = \frac{fL\bar{V}w^2}{5685D^5}. \quad . \quad . \quad . \quad . \quad . \quad (610b)$$

(c) **An example** will now be given to show the application of the preceding principles.

Find the pressure drop in 1000 ft. of 2-in. (nominal) pipe, if the fluid is water at 70 deg. Fahr. flowing with a velocity of 0.01 ft. per sec. For this case the kinematic

¹¹ Note that Eq. (608) becomes identical with Eq. (591a) (page 787) when f in (608) is equal to $4c$ in (591a). Both coefficients are often referred to as the "friction factors"; but one is just 4 times as large as the other.

viscosity, y , is found from Fig. 834 to be $1.03 (10)^{-5}$ sq. ft. per sec. Reynolds' number is

$$\frac{Dv}{y} = \frac{\frac{2.07}{12} \times 0.01}{1.03 (10)^{-5}} = 168.$$

Therefore, from Fig. 840 the ordinate $f = 0.38$; and from Eq. (606), the drop in pressure becomes

$$\Delta P = 0.38 \cdot \frac{1000}{\frac{2.07}{12}} \cdot \frac{62.3}{32.17} \cdot \frac{(0.01)^2}{2} = 0.214 \text{ lb. per sq. ft.}$$

This is a very slight pressure drop because the velocity is extremely low. For a velocity of 0.05 ft. per sec. (five times as great), the pressure drop would be 1.07 lb. per sq. ft., other conditions being the same; for viscous flow, the pressure drop increases with the first power of the velocity, as shown by Eq. (604).

(d) **Experimental determinations** of the pressure drop for viscous flow in pipes check Poiseuille's law very closely for all fluids. As soon as viscous flow is superseded by turbulent flow at higher values of Reynolds' number, the pressure drop becomes greater than that indicated by Poiseuille's law, and a new method for finding this drop in pressure becomes necessary; this method will be considered in the succeeding section.

503. Turbulent Flow of Fluids in Pipes.—(a) When the velocity of any fluid flowing through a pipe line or other close conduit becomes sufficient, turbulence will begin to appear and the viscous flow laws do not then hold. The particular value of Reynolds' number at which the transition from viscous to turbulent flow occurs is called the **critical Reynolds' number**, an exact value of which cannot be given for all cases; but from many tests made with different fluids, this value is known to vary, approximately, from 2000 to 3000, under ordinary conditions. Between these two values the region is known to be *unstable*. However, special cases have been found where the critical Reynolds' number will be much greater or less than these values.

Most of the flow problems in engineering involve the turbulent region when dealing with fluids such as air, steam, water, and gasoline; but with heavy oils and thick syrups the viscous region is also involved.

(b) The **value of the friction factor for turbulent flow** to be used in calculating the drop in pressure in a pipe can be based only on experimental data, because no one can predict the motion of the various particles of a fluid in a turbulent state. Fortunately there are now available many series of reliable tests, which enable engineers to predict

fairly well the values of the friction factor for turbulent flow under a wide variety of conditions. These data may be grouped under two headings, viz., (1) those for **smooth pipes**, such as new ones made of brass, lead, tin, or glass; and (2) those for **rough pipes** or **conduits** whose degree of roughness is difficult to determine or specify. Smooth pipes will be considered first.

(c) For **smooth pipes** the work of Stanton and Pannell¹² stands forth as very remarkable, not only for its magnitude and accuracy, but also because it shows how the principle of dimensional analysis may be applied to the problems of fluid friction. The main parts of their extensive data are shown by the curves in Fig. 841, in which Reynolds' number is plotted on a logarithmic scale as abscissa.¹³ Note that *these values of f may be used in Eqs. (606), (608), (609) and (610) for any fluid flowing in smooth pipes with turbulent or viscous flow; and that any consistent absolute system of units may be used in Eq. (606).*

From the main curve of this figure, one should note (1) that the data for water and air in various sizes of smooth pipes give a single curve with only slight deviations therefrom for all values of Reynolds' number from 3500 to 400,000; (2) that the experimentally determined points fit accurately the curve representing Poiseuille's law for values of Reynolds' number below 2000; (3) that for values of Reynolds' number lying between 2000 and 3000, roughly, the results show wide variation, indicating an *unstable* or *critical region* in which the values of the friction factor cannot be accurately predicted; and (4) that for the high values of Reynolds' number the friction factor for smooth pipes becomes very small, regardless of the pipe diameter.

For extremely low values of Reynolds' number (i.e., less than 125) Stanton and Pannell used thick oil in a steel pipe, and their results are shown by the small insert in Fig. 841.

The relation between the average and maximum velocities of a fluid in a pipe for a wide range of Reynolds' number, as obtained by Stanton and Pannell, is given by the top curve in Fig. 841; the broken part of this curve for high values of Reynolds' number merely indicates the probable values, as estimated by the authors of this text.

(d) Many **other experiments** on the flow of various fluids in **smooth pipes** have been made and the results have recently been very well

¹² "Similarity of Motion in Relation to the Surface Friction of Fluids," Proc. Royal Soc., Vol. A 214, 1914, p. 199.

¹³ In the original curve of Stanton and Pannell, the ordinate is $\left(\frac{P_1 - P_2}{v^2}\right)\left(\frac{m}{L}\right)$, which is $\frac{1}{8}$ of the friction factor in Fig. 841. Here m represents the hydraulic radius (see page 819) of the pipe or conduit.

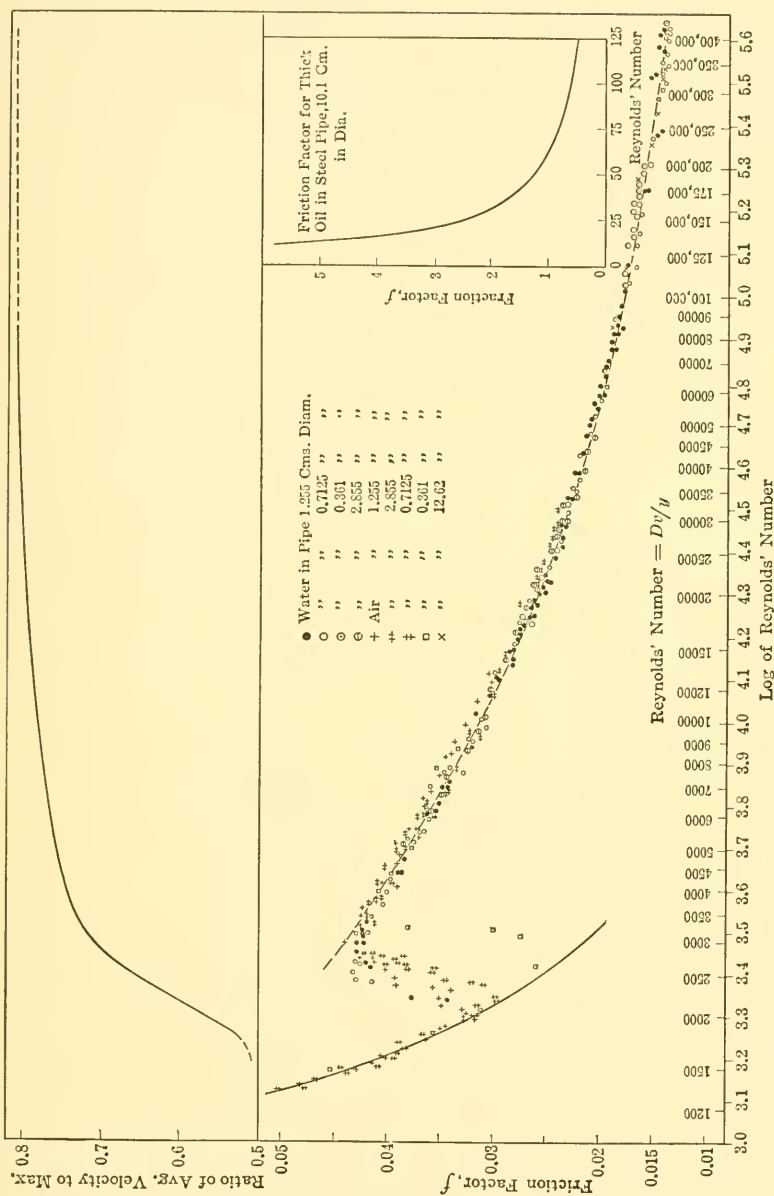


Fig. 841.—Experimental Results of Stanton and Pannell with various Fluids Flowing through Drawn Brass Pipes.

summarized by Kemler.¹⁴ He shows that all the available data on the flow of various fluids through *smooth* lead, glass, and tin pipes give nearly the same friction factor as those for brass. His results confirm those of Stanton and Pannell regarding turbulent flow in such pipes; i.e., *in smooth pipes the friction factor is independent of the size of pipe and depends only on Reynolds' number.* The values obtained from the curves in Fig. 841 may therefore be used in Eqs. (606), (608), (609), and (610) for all kinds of smooth pipe with the expectation that the results will probably be within 5 per cent of the actual ones.

(e) **Rough pipes** cannot be expected to give friction factors that are independent of the diameter *when turbulent flow is involved*, because the same size of scale or other projection on the inside surface becomes a larger fraction of the diameter the smaller the pipe, and thus the turbulence is intensified to a greater extent with the small pipe. On the other hand, with *viscous flow in rough pipes*, a thin layer of the fluid clings to the surface, and thus the roughness of such a surface does not affect the friction factor for any given value of Reynolds' number with such flow; but this number should be calculated for the real inside diameter that allows for the deposit or scale, when this deposit is known to be considerable.

When serious incrustations are expected, as in a raw-water line, the safest design for turbulent flow is to use the maximum value of f (0.054) and estimate the ultimate diameter. In steam and air lines the deposit or scale is generally very small.

The method of estimating the friction factor for various kinds of rough pipes and conduits will be given in Sect. 504 after new steel pipe lines, bends, fittings, and valves have been considered.

(f) For **new steel pipes**, which are rough relative to brass, Kemler has collected from 26 independent investigators the data for the flow of air, steam, and water, through pipes of various sizes, and then plotted these results to show the relation between friction factor and Reynolds' number from 1000 to 1,000,000. Such results show that the *extreme* values of the friction factor for all these many cases do not vary more than 10 per cent from the average as given in his summary. Although his curves included a maximum value of 10^6 for Reynolds' number, this is not sufficient to cover all the cases of modern steam power plants that use enormous quantities of steam flowing through large pipe lines at high velocities. In preparing this text special effort has, therefore, been made to secure reliable additional data regarding the flow

¹⁴ "A Study of the Data on the Flow of Fluids in Pipes," by Emory Kemler, December meeting of A.S.M.E., 1932.

through high-pressure steam lines (i.e., 400 to 1200 lb. per sq. in.). Some of these data involved Reynolds' numbers almost as high as 10^7 and flow rates approaching 1,000,000 lb. of steam per hour. The results of these steam tests fitted in well with those of Kemler, but on the whole indicated that his friction factors are slightly low in the region covered by Reynolds' numbers near 10^6 . Consequently the resultant set of friction factors that are recommended in this text for *new steel pipes* are those given in Fig. 842. The upper half of this figure gives values substantially the same as Kemler's, but redrawn to a different scale; this part of the figure also shows by two broken lines the upper and lower *limits of f for smooth pipes as given by Kemler*. The lower half of this figure contains the *curve of Stanton and Pannell for smooth pipe*; this curve is drawn as a solid line as far as their data permit and as a broken line thereafter.

The *main curves in the lower half of Fig. 842* are based largely on those of Kemler but extended to 10^7 for Reynolds' number, with values of f to agree with the data on the flow of high-pressure steam independently obtained by the authors of this text.

In this connection one should note that the values of the kinematic viscosities (ν) of high-pressure steam are not yet well known; the values of this factor used by the authors to calculate Reynolds' number for drawing the curves in Fig. 842 are those given in Fig. 835 (page 798). Consequently, if these viscosities are used to calculate Reynolds' number and then the friction factors are obtained from Fig. 842, the results will be satisfactory for high-pressure steam even though the kinematic viscosities and the friction factors may be found, eventually, to be somewhat in error.

However, the exact values of the friction factors and the viscosities are not of the utmost importance at this time. Although effort has been made to give the most reliable information available, the chief point of importance is to learn the *method* of using such data in problems involving the friction factor of any fluid. Engineers have been rather slow in adopting this method, which is, undoubtedly, the most logical and satisfactory one for general use.

(g) An **example** of the calculation of the **pressure drop in a steam line** will now be given.

Assume that a new horizontal steel pipe, having an inside diameter of 20.06 in. carries 900,000 lb. of steam per hr. If the steam enters this line with a pressure of 450 lb. per sq. in. abs. and a temperature of 700 deg. fahr., find the drop in pressure due to 50 ft. of straight pipe. Neglect heat transfer from the line.

For this case the specific volume, initially, is found to be $\bar{V}_1 = 1.455$ cu. ft.

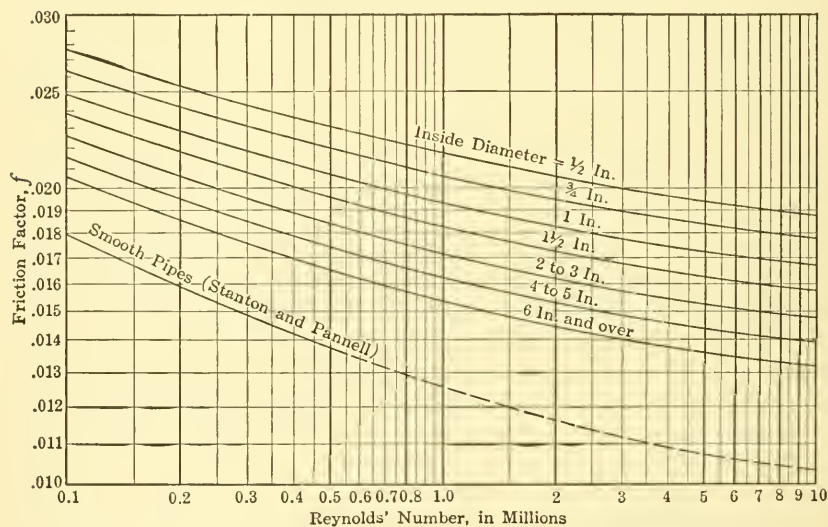
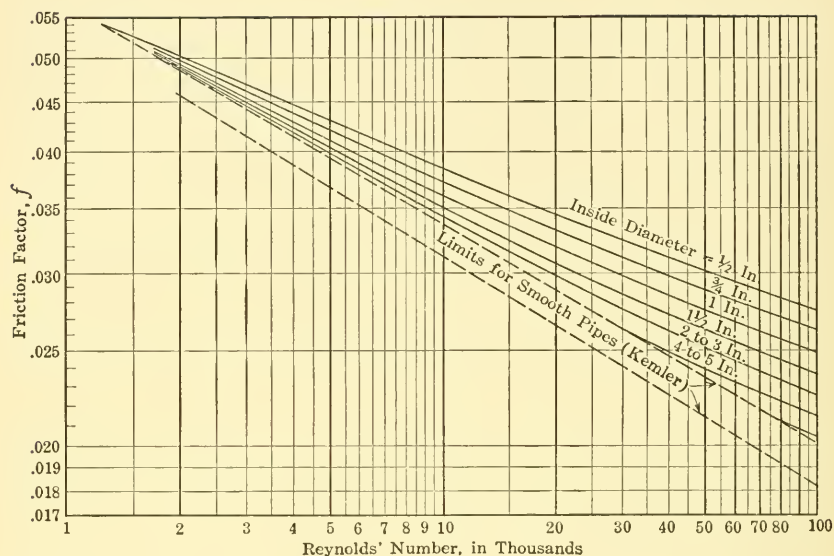


FIG. 842.—Friction Factors for the Flow of Any Fluid in New Steel Pipes.

per lb., and the rate of flow $w = 900,000 \div 3600 = 250$ lb. per sec. Hence the initial velocity $= v_1 = w\bar{V}_1/\text{area}$

$$= \frac{250 \times 1.455}{0.7854 \times \left(\frac{20.06}{12}\right)^2} = 166 \text{ ft. per sec.}$$

From Fig. 835, the kinematic viscosity (y) is found to be $3.55 (10)^{-6}$ sq. ft. per sec.

Hence Reynolds' number (Dv/y) is $\left(\frac{20.06}{12}\right) (166) \div \frac{(10)^5}{3.55} = 7.8 (10)^6$. Then, with

this number and a diameter of pipe greater than 6 in., the friction factor, f , is found from Fig. 842 to be 0.0133. Hence, from Eq. (609),

$$\Delta p = \frac{fLv^2}{772d\bar{V}} = \frac{0.0133 \times 50 \times (166)^2}{772 \times 20.06 \times 1.455} = 0.81 \text{ lb. per sq. in.}$$

or, from Eq. (610),

$$\Delta p = \frac{43.7fL\bar{V}w^2}{d^5}$$

$$= \frac{43.7 \times 0.0133 \times 50 \times 1.455 \times (250)^2}{(20.06)^5, \text{ or } 3,248,000} = 0.81 \text{ lb. per sq. in.}$$

This result agrees closely with the best available test data.

For a pressure drop as small as this, the *average* specific volume and the *average* velocity in the pipe line are nearly the same as the initial ones; hence no further calculation is needed, but for larger pressure drops a second calculation might be desirable to allow for the increased volume and velocity. When a long line is involved, or when the insulation is poor, the drop in temperature in the line as well as the drop in pressure also need to be considered in finding the *average* specific volume and velocity of a hot elastic fluid, such as steam.

504. The Flow of Fluids through Bends, Fittings, and Valves.—

(a) In pipe lines, numerous bends, fittings, and valves are likely to be needed, and the drop in pressure of the fluid in passing through them may be an important one. For *short pipe lines* the drop through the fittings and valves is generally larger than that through the pipe itself, but with *long pipe lines* the pipe alone is the chief factor in the total drop. A single satisfactory equation for estimating the pressure drop through all forms of valves and fittings has not been developed because this drop is dependent upon the magnitude of the turbulence produced in each case; this turbulence is chiefly determined by the amount of **sudden expansion and contraction**, which varies considerably with different valves and fittings. For **smooth pipe bends**, however, the estimates can be fairly well made. In all cases, whether bends, fittings, or

valves are involved, the most satisfactory estimate is made by using the **equivalent length** of straight pipe in Eqs. (606), (608) to (610), inclusive, to determine the drop in pressure.

(b) For **pipe bends**, the following equation ¹⁵ will be found satisfactory to determine the *equivalent length* (L_c) of straight pipe:

$$L_c = 12.85 \frac{l_c}{r^{0.83}}, \quad (611)$$

where l_c = length of curve measured along the center line of the pipe, and r = ratio of radius of curve on the center line to the radius of pipe.

In this equation any single convenient linear unit may be used for l_c , and then L_c will be in the same unit; but L_c must be in feet when used in Eqs. (608) to (610) inclusive. When $r = 4.5$ to 5 in Eq. (611), the resistance due to the pipe bend will generally be found to be a minimum, because greater values of r will cause the increased length of the bend to more than offset the greater turbulence caused by a smaller value of r .

After obtaining the equivalent length of straight pipe the friction factor that should be used will depend upon the character of the inner surface of the bend. For **smooth pipe bends** obtain f from Fig. 842 if the pipe is free from incrustations; and for **corrugated pipe bends**, and corrugated straight pipes, which are sometimes used in large high-temperature steam lines to give greater flexibility, f will probably lie between 0.040 and 0.054 —values that correspond to an extremely rough surface. (For rough pipe, see Sect. 505, page 816.)

(c) For **pipe fittings and valves** there is no simple and accurate method of estimating the drop in pressure through them for all velocities and all fluids. However, the equations of Foster,¹⁶ which are very easy to apply, are given here although they are now known not to hold well in all cases. For *high-pressure steam* (say above 250 lb. per sq. in.) flowing *at high velocities* through large valves and fittings, a limited amount of evidence appears to indicate that his values should be increased from 20 to 35 per cent.¹⁷

Foster's equations, applicable for **screwed fittings and valves**, are:

$$L_c = 53.75 r_f D^{1.25} \text{ (for water)} \quad (612)$$

¹⁵ This is a slightly different form of the equation given on p. 147 of "Piping Handbook," by J. H. Walker and Sabin Crocker, McGraw-Hill Book Co.

¹⁶ See "Effect of Fittings on the Flow of Fluids through Pipe Lines," by Dean E. Foster, Trans. A.S.M.E., 1920, p. 647.

¹⁷ See "High Temperature and High Pressure Steam Lines," by B. N. Broido, Trans. A.S.M.E., 1922, p. 1219.

and

$$L_e = 43.7 r_f D^{1.2} \text{ (for air, steam, and gases)} \quad . \quad . \quad (613)$$

where

L_e = length of straight pipe, in feet, having a resistance equivalent to that of the fitting.

D = inside diameter of pipe, in feet.

r_f = resistance factor of the fitting.

Foster's values of r_f are independent of the fluid and depend only on the type of fitting or valve, as shown in Table LXIX, which also shows

TABLE LXIX

LENGTHS IN FEET OF STANDARD PIPE TO ALLOW FOR VARIOUS SCREW FITTINGS AND VALVES IN CONDUITS CARRYING STEAM, AIR, OR GAS

Nominal Pipe Size, Inches		Actual Inside Diam- eter, Inches	Form of Resistance							
			Gate Valve	Long-Sweep Elbow or on Run of Standard Tee	Medium-Sweep Elbow or on Run of Tee Reduced in Size $\frac{1}{4}$	Standard Elbow or on Run of Tee Reduced in Size $\frac{1}{2}$	Angle Valve	Close Return Bend	Tee through Side Outlet	Globe Valve
Factor of Resistance, r_f			0.25	0.33	0.42	0.67	0.90	1.00	1.33	2.00
$\frac{1}{2}$	0.622	0.031	0.41	0.52	0.84	1.12	1.25	1.66	2.50	
$\frac{3}{4}$	0.824	0.044	0.57	0.73	1.17	1.57	1.75	2.33	3.50	
1	1.049	0.057	0.77	0.98	1.57	2.11	2.34	3.11	4.68	
$1\frac{1}{4}$	1.380	0.082	1.07	1.37	2.19	2.94	3.27	4.35	6.54	
$1\frac{1}{2}$	1.610	0.098	1.29	1.64	2.63	3.52	3.92	5.21	7.84	
2	2.067	1.32	1.74	2.23	3.55	4.77	5.30	7.05	10.60	
$2\frac{1}{2}$	2.469	1.64	2.16	2.75	4.39	5.91	6.56	8.71	13.12	
3	3.068	2.13	2.81	3.59	5.72	7.69	8.54	11.40	17.08	
$3\frac{1}{2}$	3.548	2.53	3.34	4.26	6.80	9.10	10.13	13.50	20.26	
4	4.026	2.96	3.90	4.97	7.94	10.65	11.84	15.75	23.68	
$4\frac{1}{2}$	4.506	3.27	4.45	5.66	9.05	12.14	13.50	17.95	27.00	
5	5.047	3.88	5.11	6.52	10.40	13.95	15.51	20.60	31.02	
6	6.068	4.81	6.35	8.09	12.90	17.35	19.27	25.60	38.54	
7	7.023	5.75	7.59	9.66	15.40	20.70	23.02	30.60	46.08	
8	7.981	6.70	8.85	11.20	17.90	24.10	26.80	35.60	53.60	
10	10.02	8.75	11.54	14.70	23.40	31.50	35.00	46.60	70.00	
12	12.09	10.90	14.40	18.35	29.30	39.30	43.70	58.10	87.40	

the magnitude of L_e for various sizes of *standard screwed fittings* for air, steam, or gases. Note that for high-pressure steam lines, the “standard screwed fittings” are not heavy enough; hence Table LXIX cannot be used for such cases, but Eq. (613) may be employed after the proper

inside diameter of the heavy piping is determined from Tables LXX and LXXI (pages 825 and 826).

When a **flanged fitting** is used the resistance is about 75 per cent of that of the screwed fitting.

Foster’s equations do not contain any velocity term, which is now known to be a factor in determining the equivalent length of straight pipe, as is shown by the curves ¹⁸ in Fig. 843.

The particular velocities at which the equivalent lengths by Foster’s formula for water agree with these experimental data are marked with circles on the curves.

For each *standard screwed elbow* a common practice for *quick* estimating has been to allow from 30 to 50 pipe diameters as the equivalent length of straight pipe.

The experimental data ¹⁹ regarding this

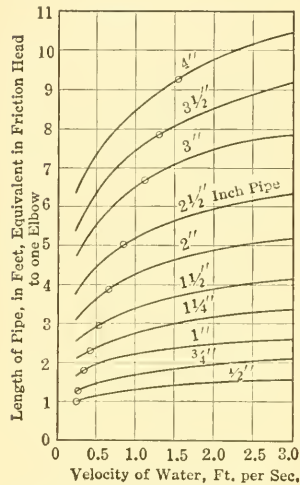


FIG. 843.—Giesecke’s Values of Equivalent Lengths of One Screwed Elbow for Water.

point indicate that an average value of 30 is fairly reliable for all *velocities above the critical*, i.e., with turbulent flow; but that in the *viscous flow region* the following values are more accurate for various values of Reynolds’ number:

Reynolds’ number.....	500	200	100	50	20
Equivalent length, in pipe diameters.....	23	10	5.5	3.5	2.5

This again serves to emphasize how important Reynolds’ number is in all problems involving the resistance to flow. Until Wilson, McAdams, and Seltzer gave their data in terms of a function similar to Reynolds’ number, the sharp break at the critical velocity in the number of diameters that represent the equivalent length was not suspected.

¹⁸ See “Discussion of Foster’s Paper,” by F. E. Giesecke, Trans. A.S.M.E., 1920, p. 663.

¹⁹ See “The Flow of Fluids through Commercial Pipe Lines,” by Wilson, McAdams, and Seltzer, Journal of Ind. and Eng. Chemistry, Feb., 1922, p. 105.

(d) With **special forms of elbows** the resistance of each one may be expressed in terms of the standard screwed ell. Thus, for an *elbow made by welding a 90-deg. miter* the resistance is from 60 to 75 per cent greater than that of the standard elbow, and for an *elbow made by two 45-deg. welds* the resistance is from 20 to 25 per cent less than that of the standard.

(e) The **proper shape of elbows for air ducts** must oftentimes be determined chiefly by the available space, but in all cases the resistance to flow must be considered. In this connection interesting experi-

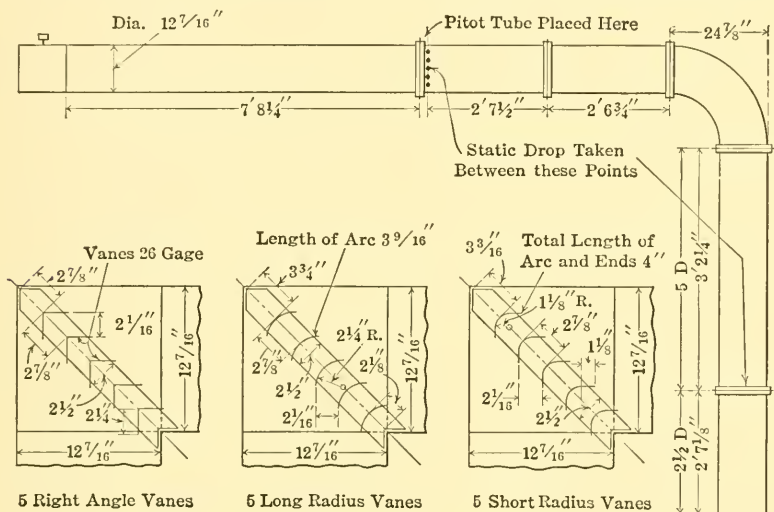


FIG. 844.—Arrangement of Apparatus and Sketch of Vane Design.

mental data are available from Payn and Stelzer, as given by Professor Brown,²⁰ from whose paper Figs. 844, 845, and 846 have been reproduced. The curves in these figures show clearly (1) that in all cases the drop in pressure is nearly proportional to the square of the velocity; (2) that the *curved elbow without vanes* causes a pressure drop of about $\frac{1}{7}$ of that of the *square elbow without vanes*; (3) that the curved vanes are far superior to the square vanes in reducing turbulence; (4) that the square elbow with curved vanes will have a resistance of about $\frac{1}{3}$ as much as the same elbow without vanes; and (5) that, when the pressure drop is expressed as a percentage of the velocity pressure, this percentage is constant for a given type of elbow, regardless of the velocity.

Regarding the **square corner elbow versus the round corner one**,

²⁰ See "Friction of Air in Elbows," by A. I. Brown, *Power Plant Engineering*, Aug. 15, 1932, p. 630.

special attention is called to the experimental data of Wirt,²¹ who showed that substituting a **square outer corner** for a *round one* in a rectangular duct will reduce the frictional resistance about 10 per cent, *provided the inner corner is rounded the same in both*. His work shows that in such cases the advantage of the square outer corner over the round one decreases as the radius of the center line of the turn is increased, and

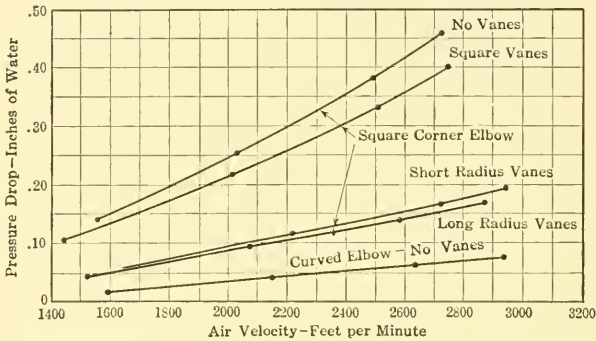


FIG. 845.—Pressure Drop for Different Velocities and Elbows.

when this radius is three or four times the diameter of the pipe the superiority disappears. Note that the *square inner corner always causes a larger loss than a rounded one*, even though the radius of the latter be small.

505. Turbulent Flow of Fluids in Rough Closed Conduits.—(a) Engineers often need to estimate the drop in pressure of fluids flowing in closed conduits that are made of various materials and in many different shapes. Naturally such problems introduce more uncertainty regarding the proper friction factor to use than is involved in the preceding cases. Even though the problem is a complex one the general principles already given and the data now available are of great assistance in making reliable estimates.

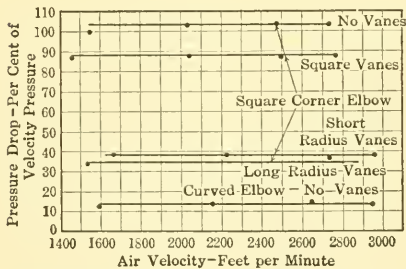


FIG. 846.—Pressure Drop Expressed in Terms of the Velocity Pressure.

(b) The **degree of roughness** of a conduit cannot be expressed precisely by merely specifying the material of which the walls are made.

²¹ See "An Experimental Investigation of Nozzle Efficiency," by H. Loring Wirt, Trans. A.S.M.E., 1924, p. 981; also, by the same author, "New Data for the Design of Elbows in Duct Systems," Gen. Elec. Rev., June, 1927, p. 286.

However, certain materials are known to be very rough relative to others; and recently the excellent work of Pigott²² has become available; he has presented some very logical data on which to make allowances for roughness. He reasoned that "if the roughness of the surface had the effect of entangling a thicker film of fluid at very low speeds near the wall, *the result would be much the same as reducing the diameter.*" Then, since the pressure loss is dependent upon the 5th power of the diameter, for any given rate of flow (see Eq. 610), a very thin scale in the pipe would affect the coefficient appreciably if the diameter is small, but not very much if the diameter is large. For example, a 1-in. I.D. steel pipe having no scale would have a net diameter 1.053 times that of the same pipe coated with a scale of 0.025 in.; but $(1.053)^5 = 1.295$; hence for such a case the rough pipe might be expected to have a value of f about 30 per cent greater than that for the smooth one. On the other hand, a 24-in. concrete pipe might have rough projections of, say, 0.15 in., which would thus cause the diameter ratio to be 24/23.7 or 1.013, and $(1.013)^5 = 1.067$; hence for this case the friction factor might reasonably be expected to be about 7 per cent more than that for a 24-in. smooth pipe. Such results as these have been checked with actual tests and the agreement is quite satisfactory; but there is no way of being sure about the exact amount of the "roughness" of any given line. However, Mr. Pigott has made estimates for the various sizes and materials ordinarily used, and by the method just given, calculated the *probable value* of the friction factor. His results are presented by the table and 18 curves given in Fig. 847. To use these curves, the material and diameter of the conduit are supposed to be known, and then the corresponding curve number may be selected directly from the table accompanying these curves, after which the value of f may be read for the known value of Reynolds' number. For example, for a well-finished cement conduit 36 in. in diameter, curve 4 would be used; but for a spiral riveted pipe of the same diameter, curve 6 would be chosen.

(c) As an **example** of the application of some of the previous equations and figures, consider the flow of air at a pressure of 14.7 lb. per sq. in. abs., through a galvanized iron pipe 18 in. in diameter and 200 ft. long.

(a) Find the rate of flow in pounds per second, and the drop in pressure in this line, when the velocity of the air is 80 ft. per sec. and the temperature is 70 deg. fahr.

(b) The same as (a) except that the temperature is 400 deg. fahr.

²² See "The Flow of Fluids in Closed Conduits," by R. J. S. Pigott, in a paper presented before the A.S.M.E. at Buffalo, June 1932; Mech. Eng., Aug. 1933, p. 497.

Solution:

Quantity	Symbol	Part (a)	Part (b)
Diameter of line, in in.	d	18	18
Diameter of line, in ft.	D	1.5	1.5
Average absolute pressure of the air, lb. per sq. in.	p	14.7	14.7
Average temperature of the air, deg. fahr.	t	70	400
Average absolute temperature of the air, deg. fahr.	T	530	860
Average specific volume, cu. ft. per lb., = $53.3T/144p$	$\bar{v}$	13.36	21.7
Rate of flow, lb. per sec., $\pi D^2 v/4\bar{v}$	w	10.56	6.51
Average velocity of the air, ft. per sec.	v	80	80
Kinematic viscosity, from Fig. 836	ν	$16 (10)^{-5}$	$38 (10)^{-5}$
Reynolds' number, Dv/ν	R_n	$7.5 (10)^5$	$3.16 (10)^5$
Friction factor, from Fig. 847, curve No. 4	f	0.016	0.0175
Pressure drop, in lb. per sq. in.	Δp	0.11	0.074
From Eq. (609), $\Delta p = fLv^2/772d \bar{v}$			
Pressure drop, in in. of water, $\Delta p/.036$	$\Delta p'$	3.06	2.06

As a check, Eq. (610b) may also be applied.

Hence, for (a)

$$\Delta p = \frac{fL\bar{v}w^2}{5685D^5} = \frac{0.016 \times 200 \times 13.36 \times (10.56)^2}{5685 \times (1.5)^5} = 0.11,$$

and for (b)

$$\Delta p = \frac{0.0175 \times 200 \times 21.7 \times (6.50)^2}{5685 \times (1.5)^5} = 0.074 \text{ lb. per sq. in.}$$

This example also serves to emphasize what is shown by Eqs. (608) to (610) inclusive; i.e., the drop in pressure, for any given fluid and conduit, is not solely dependent on the square of the velocity, but also depends directly on the density and indirectly upon the latter in so far as the viscosity is thereby affected.

(d) For **conduits having special shapes** the friction factors may vary considerably from those of circular shape. To assist in making such calculations the **mean hydraulic radius** (m) is employed instead of the diameter of a circular section. This term is defined as follows:

$$m = \frac{\text{area of fluid cross-section}}{\text{length of wetted perimeter}}. \quad . \quad . \quad . \quad . \quad (614)$$

Hence, for *circular sections*, $d = 4m$, and thus Eq. (609), page 804, would become

$$\Delta p = \frac{fLv^2}{772(4m)\bar{V}} = \frac{fLv^2}{3088m\bar{V}}; \quad \dots \quad (615)$$

and similar changes can be made in the other equations.

Many experiments have shown that for *sections that are nearly circular* the use of the mean hydraulic radius as in Eq. (615) will give satisfactory results in both the viscous and turbulent regions, provided the substitution of $4m$ for d has also been made in Eqs. (593) or (594) for finding Reynolds' number used to obtain f .

For conduits that have *sections far different from the circular*, the use of the mean hydraulic radius in Eq. (615) may give results considerably in error. The test data available indicate that for the turbulent region the error will be far less than in the viscous region, where the value of f may be as much as 50 per cent greater than it is for a circular section.

For **annular sections** formed by having one pipe of diameter d_1 surround another of diameter d_2 , the value of m becomes $\frac{1}{4}(d_1 - d_2)$; and for such a case the experimental data of Atherton²³ indicate that in the viscous region f is about 36 per cent larger than it is for a circular section, and that in the turbulent region f is about 26 per cent greater than it is for a circular section.

(e) An **upper limit** of the friction factor for the turbulent flow of a fluid in a rough conduit of any shape very probably exists; but just exactly what this limit is cannot be precisely specified yet. However, the studies of Pigott and Kemler, who examined more than 10,000 individual tests, enabled them to conclude that "unless the roughness becomes of such an order of magnitude as to cause effects such as occur when there are contractions and enlargements, the friction factor will not become greater than 0.054." Their work is based on such a large number of tests that it is very convincing; but they do not claim that this upper limit has been proved conclusively.

Note that in Fig. 847, curve 18 is drawn for its entire length with $f = 0.054$, and that all the curves have this value of f for the lower critical value of 1200 for Reynolds' number.

The upper limit of f , that would be used with the common **formula**²⁴ of Chézy, D'Arcy, or Fanning, would be $0.054/4$ or 0.0135 .

²³ "Fluid Flow in Pipes of Annular Cross-section," by D. H. Atherton, Trans. A.S.M.E., 1926, p. 145.

²⁴ This formula is widely used in hydraulic work, and is identical with Eq. (608) (page 804), except that it contains the factor 4 in the numerator; hence the friction factor in this formula is just $\frac{1}{4}$ of that used in Eqs. (606) to (612).

(f) The **transverse flow of a fluid through tube nests** is of very common occurrence in heat-transfer apparatus, such as boilers, economizers, feedwater heaters, and condensers. For such a nest of tubes the drop in pressure of a fluid with *cross-tube flow* is much more than that through a smooth conduit; but this drop may be fairly well calculated, according to Pigott, by using: (1) the upper limit of the friction factor for turbulent flow ($f = 0.054$), (2) the *true curved path* of the fluid for the length of the passage, and (3) the mean hydraulic radius based on the radial space between the tubes. This last statement means that for equal spacing of the tubes having a radial clearance space c , and a tube length l between baffles, the mean hydraulic radius would then be $\frac{cl}{2(c+l)}$, expressed in the same unit as that for c and l . This radius approaches $c/2$ as a limiting value, when l is very large relative to c , as it often is.

This maximum value of f may be used for all cases of cross-tube flow with tube diameters from $\frac{1}{2}$ to 4 in. inclusive, provided the value of the radial clearance c is not greater than the diameter of the tube, and provided the rate of flow is sufficient to cause turbulence. However, with *viscous fluids* and *low velocities* the value of Reynolds' number will be very low and may even be less than its critical value, when f will be larger than 0.054, as is shown by Eq. (607), or Fig. 840, p. 803.

Since the cross-tube flow of a fluid occurs when heat is being transferred to or from that fluid, the question naturally arises as to the effect of this heat transmission on the fluid friction. This question has been investigated very recently by Sieder and Scott,²⁵ whose results show that in the *turbulent region* the friction factor is substantially the same whether the fluid is being heated or cooled; but that in the viscous region the friction factor is more than 100 per cent greater when the fluid is being cooled than when being heated, provided the value of Reynolds' number is the same in both cases and is very small, i.e., less than 150.

506. Empirical Formulas.—(a) The **drop in pressure** of a fluid through a pipe line may, in some cases, be satisfactorily determined by means of an empirical formula involving constant coefficients that are based on a limited range of experiments; but from the previous discussion it is apparent that such a formula cannot be expected to give very accurate results when the surface of the pipe line, the fluid pressure, and the temperature are far different from those prevailing in the

²⁵ See "Fluid Friction at Parallel and Right Angles to Tubes and Tube Bundles," in a paper presented before the Buffalo Section of the A.S.M.E., June, 1932.

experiments on which the coefficients were based. For these reasons the preceding methods are recommended as being more satisfactory, in general, now that wide variations in the viscosities and densities of various fluids are encountered in many classes of engineering work; but some of these empirical formulas will also be given, chiefly for reference.

(b) The **Unwin-Babcock formula**, which has been very commonly used for the flow of steam in clean steel pipe, is as follows:

$$\Delta p = 0.000131 \left(1 + \frac{3.6}{d} \right) \frac{L \bar{V} w_m^2}{d^5}, \quad . \quad . \quad . \quad (616)$$

where d = inside diameter of the pipe, in in.

L = length of the pipe, in ft.

w_m = rate of steam flow, in lb. per min.

$\bar{V}$ = average specific volume of the steam, in cu. ft. per lb.

Δp = the drop in pressure, in lb. per sq. in.

For most cases, it is more convenient to use the flow rate, w , in *lb. per sec.*; then Eq. (616) becomes

$$\Delta p = 0.4716 \left(1 + \frac{3.6}{d} \right) \frac{L \bar{V} w^2}{d^5}. \quad . \quad . \quad . \quad (617)$$

In this form, the Unwin-Babcock formula may be directly compared²⁶ with Eq. (610a), which contains the variable friction factor, f . This factor depends on the pipe diameter and Reynolds' number $\left(\frac{Dv}{y} \right)$, as shown by the curves in Fig. 842; hence Eq. (610a) will give more accurate results over a wide range of velocities, pipe diameters, steam pressures, and temperatures than any empirical formula having a constant coefficient, such as the one under consideration. The Unwin-Babcock formula was never intended for the high steam pressures and temperatures that are often used at the present time, nor for large pipes, and tests show that it gives results that are appreciably in error in such cases.

(c) The **Fritzsche formula** has also been largely used to find the drop in pressure of air and steam in pipe lines. This formula is

$$\Delta p = \frac{0.82 L \bar{V} w^{1.85}}{d^{4.97}}, \quad . \quad . \quad . \quad . \quad (618)$$

where each symbol has the same meaning as in Eq. (617).

²⁶ See page 804.

This empirical formula is but little, if any, more convenient to use than the more accurate ones given by Eqs. (609) and (610).

507. Suitable Velocities.—(a) In all the preceding discussion regarding the flow of a fluid through any kind of a closed conduit, the velocity of the fluid has been a most important item in the calculation of the drop in pressure. Naturally, the question arises as to what velocity should be used in design work in any specific case. The correct answer can be determined only by balancing the extra cost of the larger diameter of pipe against the advantages of a smaller drop in pressure; these advantages naturally depend on the fluid and the reason why it is being delivered through the pipe line. For example, in a steam or air line in which the fluid is *to operate an engine*, a large drop in pressure is serious because it will reduce the available mechanical energy of the fluid very materially; but a very large drop may be justified in the main line of a *heating system* by using enormous velocities in a relatively small inexpensive main, provided the initial pressure is sufficiently high.

(b) **Reasonable velocities** of water and steam in pipe lines for various classes of service are given by Crocker²⁷ as follows:

Fluid	Gage Pressure, Lb. per Sq. In.	Use	Reasonable Velocities, Ft. per Min.
Water.....	25 to 40	City water.....	120 to 300
Water.....	50 to 150	General service.....	300 to 600
Water.....	150	Boiler feed.....	600
Saturated steam.....	0 to 15	Heating.....	4,000 to 6,000
Saturated steam.....	50 up	Miscellaneous.....	6,000 to 8,000
Superheated steam....	200 up	Large turbines and boiler leads.....	10,000 to 15,000

With *reciprocating engines* the flow is slightly intermittent; hence, the steam velocity is generally kept below 8000 ft. per min. For *heating mains* in which superheated steam is flowing, the velocity may be as high as 50,000 to 75,000 ft. per min.

In general, extremely high velocities may be feasible with large straight lines carrying superheated steam with a steady flow; but an intermittent flow, small lines, and wet steam all combine to require a much smaller velocity.

²⁷ See "How to Lay Out Power Plant Piping," by Sabin Crocker, *Power*, June 6, 1926, p. 890.

508. Standards for Pipe and Fittings.—(a) Standards for the materials and dimensions for pipe and pipe fittings in common use are naturally very desirable from the standpoints of both the users and manufacturers of such equipment, and there are now available a number of such standards. A very brief consideration will be given to some of the standards for piping commonly used in steam plants.²⁸

(b) **Standard-weight, extra-strong, and double-extra-strong pipe** of steel or wrought iron, such as is in common use for ordinary purposes and is known as **Commercial Wrought Pipe**, is covered as to materials, dimensions, and testing by the Standard Specifications for Welded and Seamless Steel Pipe (Series Designations A53 and A120) or by the Standard Specifications for Welded Wrought Iron Pipe (Designation A72) of the American Society for Testing Materials (A.S.T.M.). The size of such pipe is specified by the **nominal inside diameter** in the sizes ranging from $\frac{1}{8}$ -in. up to and including 12-in. The outside diameter of a given nominal size of pipe is fixed by the standard diameter of the die through which the pipe is drawn; hence, the *actual inside diameter* of that size depends upon the wall thickness and is greater with the standard weight than with the heavier weights. Above the 12-in. size the pipe is designated by the actual outside diameter and hence is known as **O.D. pipe**. Data relating to the dimensions and weight of *steel* pipe 12-in. and less falling under the different weight classifications are given in Table LXX, to which have been added the standard hydrostatic test pressures to which the pipe should be subjected before leaving the factory. A.S.T.M. *wrought iron* pipe is a little thicker than the steel pipe, hence its internal diameter is slightly less than that given in this table.

Tests on commercial pipe have shown average **bursting stresses**, in pounds per square inch, about as follows: butt-welded steel, 41,000; lap-welded steel, 52,000; seamless steel, 62,000; butt-welded wrought-iron, 29,000; lap-welded wrought-iron, 31,000. Standard-weight pipe is quite commonly limited to pressures below 100 lb. per sq. in. when used with saturated steam; and extra heavy pipe is used up to 200 lb. per sq. in., or when the pipe may be exposed to corrosion, vibration, or shock, or may have weakness at threaded joints; but higher pressures are often allowed when the conditions are very favorable, especially with pipe of quite small size. The maximum pressure allowable with a pipe of given size and thickness, or the minimum thickness for a given size and pressure, may be determined by the method given in Sect. 509(f), p. 830.

²⁸ For more detailed information regarding these standards consult the references given.

TABLE LXX
WELDED AND SEAMLESS STEEL PIPE DATA

Weight Classification	Nominal Size		Inside Diameter	External Diameter	Nominal Thickness	Transverse Areas		Length of Pipe per Sq. Ft. of External Surface	Nominal Weight per Ft.*	Hydrostatic Test Pressure	
						Internal	Metal			Butt-weld	Lap-weld and Seamless
	Inches	M.M.	Inches	Inches	Inches	Sq. In.	Sq. In.	Feet	Pounds	Lb. per Sq. In.	Lb. per Sq. In.
"STANDARD WEIGHT" PIPE	1 1/8	3	0.269	0.405	0.068	0.057	0.072	9.431	0.245	700	
	1 1/4	6	0.364	0.540	0.088	0.104	0.125	7.073	0.425	700	
	3/8	10	0.493	0.675	0.091	0.191	0.167	5.658	0.568	700	
	1 1/2	13	0.622	0.840	0.109	0.304	0.250	4.547	0.852	700	
	3/4	19	0.824	1.050	0.113	0.533	0.333	3.637	1.134	700	
	1	25	1.049	1.315	0.133	0.864	0.494	2.904	1.684	700	
	1 1/4	32	1.380	1.660	0.140	1.495	0.669	2.301	2.281	700	1000
	1 1/2	38	1.610	1.900	0.145	2.036	0.799	2.010	2.731	700	1000
	2	50	2.067	2.375	0.154	3.355	1.075	1.608	3.678	700	1000
	2 1/2	64	2.469	2.875	0.203	4.788	1.704	1.328	5.819	800	1000
	3	76	3.068	3.500	0.216	7.393	2.228	1.091	7.616	800	1000
	3 1/2	90	3.548	4.000	0.226	9.886	2.680	0.954	9.202		1000
	4	100	4.026	4.500	0.237	12.730	3.174	0.848	10.889		1000
	5	125	5.047	5.563	0.258	20.006	4.300	0.686	14.810		1000
	6	150	6.065	6.625	0.280	28.891	5.581	0.576	19.185		1000
	8	200	8.071	8.625	0.277	51.161	7.265	0.442	25.000		800
	8	200	7.981	8.625	0.322	50.027	8.399	0.442	28.809		1000
	10	250	10.192	10.750	0.279	81.585	9.178	0.355	32.000		600
	10	250	10.136	10.750	0.307	80.691	10.072	0.355	35.000		700
	10	250	10.020	10.750	0.365	78.855	11.908	0.355	41.132		900
"EXTRA STRONG" PIPE	12	300	12.090	12.750	0.330	114.800	12.876	0.299	45.000		600
	12	300	12.000	12.750	0.375	113.097	14.579	0.299	50.706		800
	1 1/8	3	0.215	0.405	0.095	0.036	0.093	9.431	0.314	700	
	1 1/4	6	0.302	0.540	0.119	0.072	0.157	7.073	0.535	700	
	3/8	10	0.423	0.675	0.126	0.141	0.217	5.658	0.738	700	
	1 1/2	13	0.546	0.840	0.147	0.234	0.320	4.547	1.087	700	
	3/4	19	0.742	1.050	0.154	0.433	0.433	3.637	1.473	700	
	1	25	0.957	1.315	0.179	0.719	0.639	2.904	2.171	700	
	1 1/4	32	1.278	1.660	0.191	1.283	0.881	2.301	2.996	1500	2500
	1 1/2	38	1.500	1.900	0.200	1.767	1.068	2.010	3.631	1500	2500
	2	50	1.939	2.375	0.218	2.953	1.477	1.608	5.022	1500	2500
	2 1/2	64	2.323	2.875	0.276	4.238	2.254	1.328	7.661	1500	2000
	3	76	2.990	3.500	0.300	6.605	3.016	1.091	10.252	1500	2000
	3 1/2	90	3.364	4.000	0.318	8.888	3.678	0.954	12.505		2000
	4	100	3.826	4.500	0.337	11.497	4.407	0.848	14.983		2000
	5	125	4.813	5.563	0.375	18.194	6.112	0.686	20.778		1800
	6	150	5.761	6.625	0.432	26.067	8.405	0.576	28.573		1800
	8	200	7.625	8.625	0.500	45.663	12.763	0.442	43.388		1500
	10	250	9.750	10.750	0.500	74.662	16.101	0.355	54.735		1000
	12	300	11.750	12.750	0.500	108.434	19.242	0.299	65.415		1000
"DOUBLE EXTRA STRONG" PIPE	1 1/2	13	0.252	0.840	0.294	0.050	0.504	4.547	1.714	700	
	3/4	19	0.434	1.050	0.308	0.148	0.718	3.637	2.440	700	
	1	25	0.599	1.315	0.358	0.282	1.076	2.904	3.659	700	
	1 1/4	32	0.896	1.660	0.382	0.630	1.534	2.301	5.214	2200	
	1 1/2	38	1.100	1.900	0.400	0.950	1.885	2.010	6.408	2200	3000
	2	50	1.503	2.375	0.436	1.774	2.656	1.608	9.029	2200	3000
	2 1/2	64	1.771	2.875	0.552	2.464	4.028	1.328	13.695	2200	3000
	3	76	2.300	3.500	0.600	4.155	5.466	1.091	18.583	2200	3000
	3 1/2	90	2.728	4.000	0.636	5.845	6.721	0.954	22.850		2500
	4	100	3.152	4.500	0.674	7.803	8.101	0.848	27.541		2500
	5	125	4.063	5.563	0.750	12.966	11.340	0.686	38.552		2000
	6	150	4.897	6.625	0.864	18.835	15.637	0.576	53.160		2000
	8	200	6.875	8.625	0.875	37.122	21.304	0.442	72.424		2000

* These weights are for plain ends, except for the "standard weight" pipes which are threaded and coupled.

(c) **Pipe for high-temperature service**, when the temperature is between 450 and 750 deg. fahr., is usually of lap-welded or seamless steel. The A.S.T.M. Standard Specifications for such conditions and for working steam pressures (W.S.P.) of 250, 400, 600, 900 and 1350 lb. per sq. in., respectively, have the designation A 106. The dimensions, weights, and test pressures specified for such pipe are given in Table LXXI for the various pressure classifications. By using an appropriate reduction in pressure, higher temperatures than 750 deg. fahr. can be used with this pipe.²⁹ (See Sect. 509(f)).

(d) **Pipe flanges and fittings** have also been standardized by the American Engineering Standards Committee (A.E.S.C.) or its successor, the American Standards Association (A.S.A.), or by the A.S.T.M. The list of these specifications is quite extensive, and space is not available here for giving it or for discussing the standards themselves. Copies of these specifications can be obtained from the organizations mentioned or from the A.S.M.E., and abstracts of them will be found

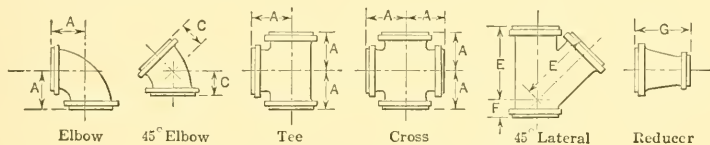


FIG. 848.—Usual Flanged Pipe Fittings.

in the more recent reference books which cover the subject of piping in detail. Specifications approved by the A.S.A. are known as *American Standard Specifications*. Usual flanged pipe fittings are shown in Fig. 848. Flanges will be discussed later (page 834). Cast-iron pipe is usually made to conform to the standards of the American Water Works Association.

509. Steam Plant Piping.—(a) The modern steam power plant contains a wide variety of piping which requires very careful planning. Limitations of space, however, permit including in this text only a very brief outline of some of the main factors involved. For a more complete study one should consult reference books, codes of piping standards and specifications, and the periodical literature dealing specially with this subject.³⁰

²⁹ It is probable that a new system for high-temperature service will be established in the near future.

³⁰ References: The several bulletins relating to piping standards and to pipe flanges and fittings as approved by the A.S.M.E. and the A.S.A.; the A.S.M.E. Boiler Construction Code; the Standard Specifications for Power Piping of the Power Piping Society, obtainable from the Power Piping Co., Pittsburgh, Pa.; and the Piping Handbook, by Walker and Crocker, McGraw-Hill Book Co.

(b) The **chief pipe lines** needed in a steam power plant are: (1) the *main steam lines* going from the steam generators to the prime movers; (2) *auxiliary lines* going to steam-driven auxiliaries; (3) the *exhaust steam lines*, main or auxiliary, including the atmospheric exhaust lines; (4) *heater steam lines* leading to feedwater heaters and evaporators; (5) *boiler feedwater lines*, high pressure; (6) *water piping for the feedwater heating system*; (7) *condensing water lines*, low pressure; (8) *vacuum lines* for the condensate and air from the condenser; (9) *blowoff piping* for the boilers, and vents for safety valves and feedwater heaters; (10) *drain lines* from the main turbines, boilers, superheaters, pumps, pipe lines, and all steam-driven auxiliaries; (11) *oil piping* for lubrication. Additional piping may be required for (12) the *service-water system*; (13) *fire pumps*; (14) *fuel pipe lines*, when pulverized coal, oil, or gas is the fuel; (15) *air lines* for the compressed air used around the plant; (16) *control system lines* for air or oil used in the combustion control system; (17) *heating lines* for the heating system in the plant; and (18) there may be various other systems.

The mere naming of these systems will indicate to some extent at least the magnitude of the work involved in designing the piping system for a large plant; so complex is the problem of the layout that recourse is often had to isometric drawings and piping models.³¹

(c) The **basic requirements** of all piping systems are: (1) safety, (2) suitability, (3) reliability, and (4) reasonable cost.

(1) **Safety** depends primarily upon the use of suitable materials, pipe thicknesses, fittings, methods of attachment and support, provision for expansion, and workmanship.

(2) **Suitability** depends on the proper selection and arrangement of the piping and valves in the system so as to establish adequate circuits for the fluid flow and to provide for convenience in operation and for maintenance.

(3) **Reliability** depends on the same factors as those for safety, and also on the tightness against leakage, on the proper functioning of drains and traps, and on the arrangement of the piping system.

(4) The **cost factors** involved consist of the initial investment and the cost of operation and maintenance, and they depend on the steam pressure and temperature, size and length of the pipe, number and kind of fittings, pressure drop, and loss of heat by radiation. Piping for a steam power plant costs from \$3.80 to \$16 per kw. of plant capacity; in some cases its cost approaches that of the prime movers, and in a large plant may total several millions of dollars.

³¹ See "Models Make for Accessibility in Delray Piping," by Frank Thorp and R. M. VanDuser, Jr., in *Power*, Dec. 17, 1929, p. 956.

(d) The **principal systems of pipe arrangements** used in multiple-unit plants are: (1) the single-header, (2) the double-header, (3) the ring, and (4) the unit systems. They apply to both the main and auxiliary systems of piping.

(1) A **single-header system** contains a single large main or header to which all supply and delivery steam lines are attached, and it is usually provided with valves by means of which any part of the main can be isolated for repairs; but the cutting out of a section may interfere to a considerable extent with the operation of the plant.

(2) In the **double-header system** the steam mains, and the respective supply and delivery lines connected thereto, are in duplicate, thus ensuring great dependability but at twice the cost of the single-header system and with twice as many joints to keep tight.

(3) The **ring system** is the outgrowth of (2); in it the ends of the two mains are connected together so as to form a ring or loop, and valves are inserted in the loop in such manner that any section can be isolated without interfering with the rest of the system. Usually the supply and delivery lines connected to the loop are not in duplicate.

(4) In the **unit system** each prime mover and the complement of steam generators serving it are connected together to form an individual system, but usually the units are also interconnected by equalizer pipes, which are in effect small headers between units. Many of the modern plants use the unit system in preference to the others.

By arranging the rows of prime movers and of steam generators **back-to-back**, i.e., having parallel adjacent rows, the length of headers needed is much less than in the **end-to-end** arrangement, in which the row of prime movers is placed in line with the row of steam generators.

(e) The **materials** used in the various pipe lines in power plants include steel, wrought iron, cast steel, cast iron, brass, and copper. For low pressures and low temperatures the choice of material may depend chiefly on a consideration of cost and on the resistance of the material to corrosive action. Wrought-iron pipe should not be used with temperatures above 750 deg. fahr. or with pressures above 400 lb. per sq. in. For high temperatures (up to 750 deg. fahr.) lap-welded or seamless steel piping is commonly used for steam lines. For unusually high-temperature steam, say 1000 deg. fahr., special alloy steels, such as chromium-nickel steel, may be needed. Brass is sometimes used for low-pressure boiler feed lines; and brass or copper piping is used for oil lines and for connections to instruments. Brass pipe is not used for pressures above 300 lb. per sq. in. or temperatures above 400 deg. fahr. Cast steel is commonly used for valve bodies, fittings, and flanges for high-temperature service. Cast iron is very suitable for outside or

underground water lines or drains and for circulating water piping, but is never used with superheated steam or with steam pressures above 250 lb. per sq. in., or with temperatures above 450 deg. fahr., because of its continuous growth under repeated changes in temperature. Specifications for piping material have been provided by the A.S.T.M., to which reference has been made elsewhere.

(f) The **thickness of steam piping** for a given diameter and pressure, or the working steam pressures allowed with a given pipe, as set forth in the various standard specifications referred to elsewhere, are based upon the modified Barlow formulas and the stresses given in the A.S.M.E. Boiler Construction Code from which the following is taken from pages 9 and 10 of the 1931 edition:

Thickness of Steam Piping.—In determining the thickness to be used for pipes at different pressures and for temperatures not exceeding 750 deg. fahr. for steel or iron pipe, and 406 deg. fahr. for brass and copper pipe, the following formulas are to be used:

For pipes having nominal diameters of from $\frac{1}{4}$ in. to 5 in.,

$$p = \frac{2S}{d} (t - 0.065) - 125. \quad : : : : : (619)$$

For pipes of nominal diameter over 5 in.,

$$p = \frac{2S}{d} (t - 0.1), \quad : : . : : : : (620)$$

where p = working pressure, lb. per sq. in.

t = thickness of wall of pipe, in.

d = actual outside diameter of pipe, in.

S = 12,000 lb. per sq. in. for seamless med. carbon steel pipe

= 9,000 lb. per sq. in. for seamless low carbon steel pipe

= 7,000 lb. per sq. in. for lap-welded steel pipe

= 5,000 lb. per sq. in. for butt-welded steel pipe

= 5,300 lb. per sq. in. for lap-welded wrought iron pipe

= 4,500 lb. per sq. in. for butt-welded wrought iron pipe

= 4,500 lb. per sq. in. for brass pipe

= 4,000 lb. per sq. in. for copper pipe.

When the temperature exceeds 700 deg. fahr., the stress for steel pipe shall not exceed that given in Table LXXII.

(g) The **thermal expansion** of pipe lines carrying high-temperature fluids should be given careful consideration because it very often causes stresses many times those produced by the pressure of the fluid. Even a relatively short line may have considerable expansion, for which provision should be made. For example, a steel pipe line 100 ft. long would have its length increased about 6.33 in., or slightly more than

TABLE LXXII

WORKING STRESSES AT VARIOUS TEMPERATURES

Max. Temp., Deg. Fahr.	Minimum of Specified Range of the Tensile Strength of the Material, Lb. per Sq. In.				
	45,000	50,000	55,000	60,000	75,000
700	9,000	10,000	11,000	12,000	15,000
750	8,220	9,110	10,000	11,200	13,000
800	6,550	7,330	8,000	9,000	10,200
850	5,440	6,050	6,750	7,400	8,300
900	4,330	4,830	5,500	5,600	6,000
950	3,200	3,600	4,000	4,000	4,000

This table applies to all plain carbon steels for which specifications are given in the Code. Where the minimum tensile strength of the specified range or the temperature differs from those for which values are given in the table, the values for S for such materials may be determined by interpolation.

0.5 per cent, when heated from the temperature of 50 to 750 deg. fahr. This amount of expansion is calculated by using a *mean linear coefficient of expansion* of $7.62 (10)^{-6}$ per deg. fahr.,³² which is probably a very reliable value for *low-carbon steel*. The coefficient of expansion, however, varies appreciably with the temperature range. For example, Moehel gives for the steel previously mentioned the following values for the different temperature ranges:

Range of temperature, deg. fahr.	32 to 212	32 to 390	32 to 570	32 to 750	32 to 930
Mean coefficient $\times 10^6$, per deg. fahr.	6.66	7.00	7.35	7.61	7.94.

The expansion coefficient also varies appreciably with the composition of the material.

For *wrought-iron pipe* the mean coefficient of expansion is almost 5 per cent greater than that for steel in the same temperature range;

³² This is the *mean value* for open-hearth steel having 0.2 to 0.3 per cent carbon and less than 0.04 per cent phosphorus, for a range of temperature from 50 to 750 deg. fahr. See the "Thermal Expansion of Metals," by Norman L. Moehel in the "Symposium on Effect of Temperature on the Properties of Metals," 1931, p. 531, issued jointly by the A.S.T.M. and A.S.M.E.

A test at the Trenton Channel Plant of the Detroit Edison Co. gave an average coefficient of $7.57 (10)^{-6}$ per deg. fahr. for steel pipe heated from 77 to 689 deg. fahr., as reported in Power, Aug. 3, 1926, p. 173.

for *copper* the coefficient is about 40 per cent greater than that of steel when the temperature rise is from 0 to 600 deg. fahr. and about 44 per cent more for a range from 0 to 400 deg. fahr.; for *cast iron* the coefficient is from 9 to 12 per cent less than that for steel.

(h) **Provision for expansion** of pipe lines is made by using plain,

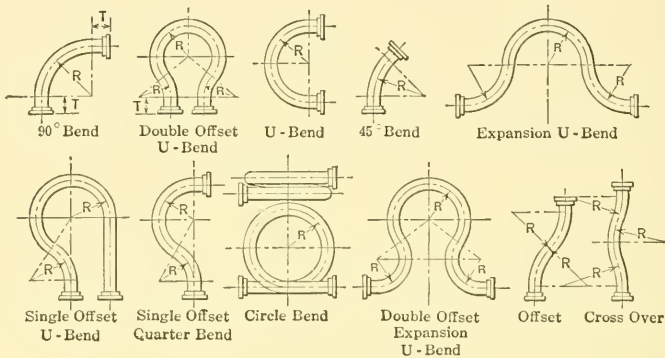


FIG. 849.—Standard Pipe Bends.

creased, or corrugated flexible bends (Figs. 849 and 850), by having offsets or changes in direction in the pipe line itself, or by introducing in the lines expansion joints of either the slip-sleeve (Fig. 851), swivel, or corrugated (Fig. 852) types. Corrugations are reinforced by means of

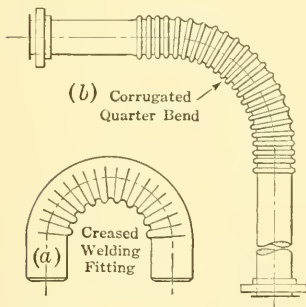


FIG. 850.—Creased and Corrugated Bends.

heavy rings when used with the higher steam pressures. Expansion joints for turbine exhaust piping are discussed on page 897. The combined stress in steel pipe due to internal pressure, thermal expansion, and dead weight between supports should not exceed 15,000 lb. per sq. in. for temperatures below 700 deg. fahr., and must be reduced for higher temperatures. It is common practice to cut pipe lengths short by an amount equal to half the thermal expansion and to cold-spring the pipe into place during the erection. The

amount of cold-springing is not, however, credited as reducing the calculated stress due to thermal expansion. Recently much study has been given to the details of pipe bends of various types and to the determination of load-deflection data for them, and the results, which have been published in several papers, should be studied when design-

ing large steam lines.³³ Pipe lines should be rigidly anchored at suitable points and should be adequately supported and guided.

(i) **Pipe connections** used with ordinary underground *cast-iron water or gas pipe* are commonly of the somewhat flexible *bell and spigot type* (Fig. 853), either with leaded and calked joints or with cemented

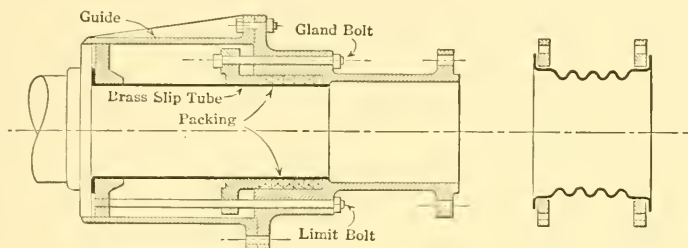


FIG. 851.—Sleeve Joint.

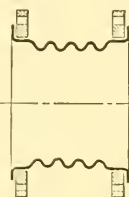


FIG. 852.—Corrugated Joint.

joints, but joints of the *swivel type* with articulating spherical surfaces are used when great flexibility is necessary, and bolted *flanged joints* are employed when stronger and tighter connections are required and the pipe is well supported. Lengths of *wrought-iron or steel pipe* may be connected by means of threaded (sleeve) *couplings* (Fig. 854), or *unions* (Fig. 855), by *bolted flanges*, or by *welding*.

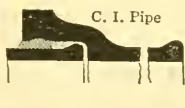


FIG. 853.—Bell and Spigot Coupling.



FIG. 854.—Sleeve Coupling.

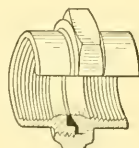


FIG. 855.—Union.

³³ See "The Elasticity of Pipe Bends," by Sabine Crocker and Sterling S. Sanford, Trans. A.S.M.E., 1922, p. 547; "Design of Steam Piping to Care for Expansion," by W. H. Shipman, Trans. A.S.M.E., FSP-51-52, 1929, p. 415; "Stresses in Turbine Pipe Bends," by A. M. Wahl, J. W. Bowley, and G. Back, Mech. Eng., Nov. 1929, p. 823; "A Graphic Method for Determining Expansion Stresses in Pipe Lines," by C. T. Mitchell, Trans. A.S.M.E., FSP-52-25, 1930, p. 167; "Steam Piping for 1000 Deg.," by J. H. Walker, Power Plant Eng., June 1, 1931, p. 606; "Load Deflection Relations for Large Plain, Corrugated, and Creased Pipe Bends," by E. T. Cope and E. A. Wert, Trans. A.S.M.E., FSP-54-12, 1932, p. 115; "Pipe Design Made Easy by a New System of Charts," by E. A. Wert, S. Smith and E. T. Cope, Power, April, 1933, p. 198; and "The Design of Piping for Flexibility by the Use of Graphs," by the last mentioned authors, presented at the Chicago Meeting of the A.S.M.E., June, 1933.

Flanges may be integral with the pipe, as when they are part of a cast pipe or when they are welded to the pipe, or they may be loose. Loose flanges are of cast iron, semi-steel, cast steel, forged steel, or rolled steel, as the conditions require. The usual forms of such flanges



FIG. 856.



FIG. 857.

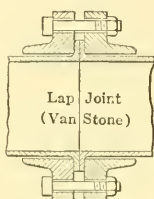


FIG. 858.



FIG. 859.

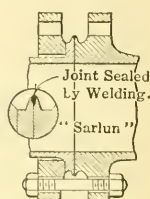


FIG. 860.

include the screwed or *threaded* type (Fig. 856), the *shrunk and peened* type (Fig. 857), the *lap joint* or *VanStone joint* (Fig. 858), the "*Midwest*" (Fig. 859) with spherical contact surface, and the patented "*Sarfun*" (or modified "*Sargol*") with joint sealed by welding (Fig.

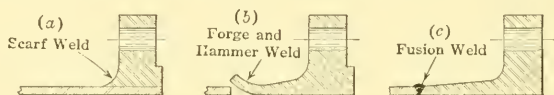


FIG. 861.—Several Kinds of Welded Flanges.

860). Several kinds of welded flanges are shown in Fig. 861. Pipe flanges and their threads and bolts should conform to the American Standard Specifications.

Flange facings of various kinds, which have been arranged for the proper retention of gaskets, and for which the dimensions have been

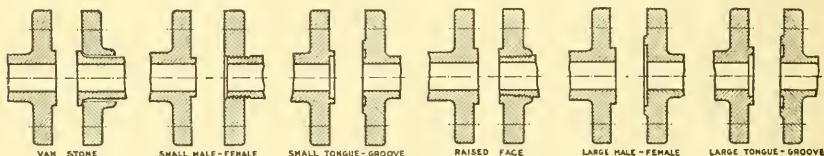


FIG. 862.—Standard Flange Facings.

standardized, are shown in Fig. 862. **Gaskets** for low pressures and temperatures are often of rubber-compounded sheet packing or of corrugated copper. Lead-alloy gaskets are employed on hydraulic and oil lines. On steam lines, gaskets having an asbestos base are satisfactory except in lines which are alternately hot and cold; soft copper or aluminum rings are sometimes used in tongue-and-grooved joints;

but for high-pressure high-temperature (750 deg. fahr.) conditions a narrow copper-coated steel ring is found to be very satisfactory.³⁴

(j) **Welding** is now used extensively in fabricating pipe lines for steam plants. Joints made by this method remain permanently tight and their employment reduces the number of screwed and flanged connections and fittings, decreases weights, lowers the cost of the piping and its erection and maintenance, minimizes the radiation losses and cost of installation, and often enables making better piping designs and simpler layouts than are

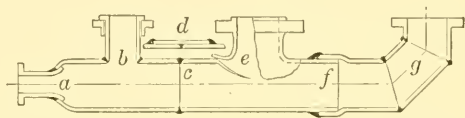


FIG. S63.—Header with Various Types of Welds.

possible when threaded or flanged joints and standard fittings are used. Fig. S63 shows a special header fabricated by using various types of welded joints; and Figs.

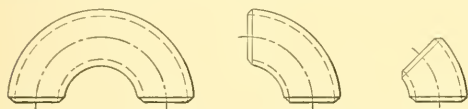


FIG. S64.—Plain Welding Bends.

S64 and S50(a), respectively, show commercial forms of plain and creased welding bends.

(k) **Valves** of the ordinary kinds are of two general types: (1) globe or angle (Fig. S65), and (2) gate (Fig. S66). *Globe or angle valves* are always used when throttling is necessary. The angle valve is in effect

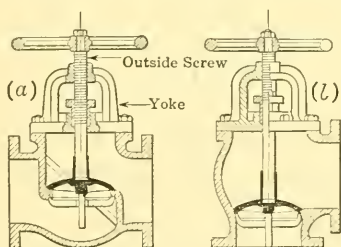


FIG. S65.—Globe and Angle Valves.
("O. S. & Y.")

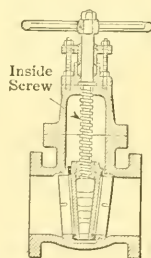


FIG. S66.—Gate Valve, with
Inside Screw.

a combination of a globe valve and an elbow, and it eliminates the need of the latter. Where suitable, the *gate valve* should be used because

³⁴ See "Keeping Steel Gaskets Tight at 1350 Lb.," by W. P. Saunier, in *Power*, July 1, 1930, p. 22. Also see "Steam Piping for 1000 Deg.," by J. H. Walker, in *Power Plant Eng.*, June 1, 1931, p. 606, or in "Heating, Piping and Air Conditioning," April 1931, p. 283; and "Leak-Tight Joints—600 to 550,000 Lb.," by F. E. Wertheim, in *Heating, Piping and Air Conditioning*, July and August 1932, pp. 469 and 546. This last article contains a bibliography on the subject.

it offers practically no obstruction to flow when open; but it should be used only wide open or fully closed. Valve bodies are made of cast iron, malleable iron, semi-steel, cast steel, forged steel, brass, bronze, or special alloys, and may have threaded, flanged or welding coupling-ends according to requirements. Valve stems and discs must be of non-corrosive and non-erosive material, and the stem threads may be external or internal with reference to the valve body. When used with high-pressure and high-temperature steam the threaded connections are not used in the larger sizes (e.g., 2 in. and above, with steam pressures 250 lb. per sq. in. and temperature 450 deg. fahr.), and the valves must be of the outside-screw-and-yoke type ("O. S. & Y."). High-pressure valves of the larger sizes have small by-pass valves which are used for equalizing the pressure on the two sides of the valve discs before they are opened. Valves may be operated by hand or by remotely controlled motors.

Other forms of valves include cocks, check valves, reducing valves, atmospheric relief valves (see Fig. S98, page 882), and the boiler valves (safety, page 733; blowoff, page 737; and non-return, which is a combination of check and stop valve) which should conform to the Boiler Construction Code (A.S.M.E.).

When laying out piping, reference should be made to the forthcoming American Standard Codes for Pressure Piping and Fabrication Details, and to the latest revision of the Standard Specifications for Piping of the Power Piping Society.

CHAPTER XLIII

THE DRAFT SYSTEM

510. Definitions of Draft.—The term **draft** is used in a large number of ways in the English language; and engineers are not in agreement regarding the meaning of the term even in its technical application to the flow of gases under low pressures. Thus the following definitions are in use:

- (1) *Draft* is the *difference* in the absolute pressure of the gas at any given point in a closed conduit and that of the ambient atmospheric pressure at the same elevation. With this definition the draft is *positive* when the gas pressure is greater than atmospheric, and *negative* when less; hence this definition permits the term “forced-draft” to be consistently used.
- (2) *Draft* is the *excess* of atmospheric pressure over the absolute gas pressure (both measured simultaneously at the same elevation) provided the gas pressure is *less* than atmospheric. With this definition the draft is always positive, and when gas pressures greater than atmospheric are involved the term *pressure* is applied instead of *draft* to indicate the *excess* of gas pressure over atmospheric, and the term “forced-draft” is hardly appropriate.
- (3) *Draft* is the *difference* in the absolute *pressures of the same gas* at two points in a closed conduit between which the gas tends to flow.
- (4) *Draft* is the *quantity* of air flowing through a furnace.
- (5) *Draft* is the *velocity* of the gas itself.

The fifth definition is not to be considered of any importance in power-plant work, but is merely a layman’s conception of the term as he applies it to the flow of air past his body in a building. The fourth definition is used very little and is not recommended. On the other hand, definitions (1), (2), and (3) are in common use in the literature pertaining to power-plant operation, and of these three, the authors’ preference is for the first one; but when reading the literature on the subject one must generally be prepared to determine the meaning of the term from the context.

511. Applications of Draft.—(a) The **draft system** in a steam-generating plant consists of (1) the air passages supplying air to the burning fuel, the passages through the fuel bed (if any), and the gas passages through the furnace, heat-absorbing apparatus, flues, dust catchers, and chimney; (2) the means for supplying the air at the

required rate; (3) the means for producing the flow of gases through their passages at the proper rates; and (4) the means for discharging the gases at such an altitude that they will not be objectionable or injurious to life or to inanimate objects in the neighborhood.

(b) The **apparatus used to produce draft** is built in a variety of forms from which a choice must be made for installation in any particular plant. **Chimneys**¹ are almost invariably used for discharging the gases at a high elevation and for producing at least part of the draft required in the system, this draft being termed either **chimney draft** or **natural draft**. When the natural draft is not sufficient to meet the full requirements, it must be supplemented by **artificial draft**, which is usually produced by mechanical means (e.g., by fans) and is then called **mechanical draft**. Sometimes the artificial draft is produced by steam jets, as in the conventional steam locomotive; sometimes it is produced by an air jet at the throat of a Venturi stack. In steam power plants having natural or mechanical draft, steam-jet blowers are sometimes used for emergencies and for peak-load operation. In a **forced-draft system** the combustion air is supplied to the furnace at a pressure above atmospheric by means of a forced-draft fan (or by a steam-jet air injector). In an **induced-draft system** an induced-draft fan (or a steam-jet or air-jet exhaustor) is placed between the steam-generator outlet and the chimney, and is used to aid the natural draft in producing and maintaining the necessary rate of air and gas flow through the system. In a **balanced-draft system** forced draft is used in combination with natural or induced draft in such a manner that the pressure in the combustion space of the furnace is nominally atmospheric (actually the pressure is maintained slightly lower for the reasons given on pp. 600 and 601).

(c) The **function of the draft-producing equipment** is to furnish the energy necessary to maintain the required flow of air and gas through the system. This energy may come from fans or jets, or it may be derived from the stack gases which would then have to be hotter than they would need to be if the same draft were produced by mechanical means. Always, however, the energy supplied must be sufficient to accelerate the gases, to overcome the fluid friction encountered by the gas stream, including that through the fuel bed, and, in many cases, to raise the masses of gas from lower to higher elevations. This is a problem in the flow of fluids to which the principles set forth in the preceding chapter apply.

When the energy for producing the draft is derived from the stack

¹ "The Boiler House at Princeton Has No Stack," by Arthur M. Greene, Jr., *Power*, Sept. 14, 1926, p. 390.

gases the operation depends upon the temperature difference maintained between the column of ascending gas within the chimney and the outside atmosphere. This action is known as the *stack effect* and will be taken up in detail in the following section. Note that the stack effect does *not* depend upon a temperature *drop* within the chimney itself, but upon the smaller average density of the stack gases as compared with that of the atmosphere.

512. Natural Draft.—(a) Consider a **column of hot gases** in a chimney, or stack, having the height Z_c feet above the furnace grate, and assume that a flexible air-tight membrane, M , is fastened over the grate surface as shown in Fig. 867, so that *no flow* of air can take place. Then if the chimney gases have an average temperature, T_c , higher than atmospheric, the membrane will have to sustain a resultant difference in pressure between its under and upper sides that will tend to make it bulge upward as indicated in the figure. This difference in pressure may very properly be called the **static draft**² of the chimney; it is also called the “stack or chimney effect” and is due simply to the difference between the density of the column of atmospheric air of height Z_c and that of the column of gases of the same height inside the chimney. The atmospheric pressure P_A at the top of the stack acts on both gas columns and does not therefore affect the draft except in so far as it affects the average density of each column.

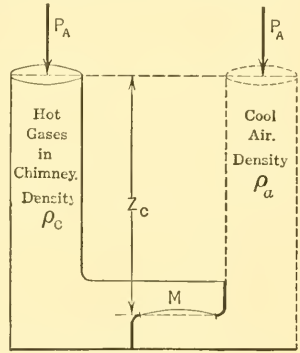


FIG. 867.

(b) The **calculation of the static draft** produced by a stack or chimney is very simple. Thus if the height is Z_c feet, and the average density of the chimney gas is ρ_c pounds per cubic foot while that of the ambient atmosphere is ρ_a , then the static draft, in *pounds per square foot*, is

$$\Delta P = Z_c (\rho_a - \rho_c). \quad (621)$$

Or, since 1 in. of water at 60 deg. fahr. corresponds to a pressure of 5.2 lb. per sq. ft. or 0.0361 lb. per sq. in., this draft, in *inches of water*, is

$$\Delta p' = \frac{Z_c}{5.2} (\rho_a - \rho_c). \quad (622)$$

But the *average* densities of the two gas columns are conveniently expressed in terms of their *average* absolute temperatures (T_a and T_c),

² The term *theoretical draft* is also used; but it is far less satisfactory because it is less significant.

their absolute pressures (p_a and p_c), and the two gas constants (R_a and R_c); hence, since $144/5.2 = 27.7$, Eq. (622) becomes

$$\Delta p' = 27.7 Z_c \left(\frac{p_a}{R_a T_a} - \frac{p_c}{R_c T_c} \right). \quad . \quad . \quad . \quad . \quad (623)$$

The value of R_a is 53.3, while that of R_c depends upon the composition of the stack gases. Typical values of R_c for different fuels burned with various amounts of excess air are approximately as given in Table LXXIII.

TABLE LXXIII
VALUES OF THE CHIMNEY GAS CONSTANT, R_c

Fuel	Excess Air, Per Cent			
	0	25	50	100
Dry wood.....	52.4	52.6	52.7	53.0
Pennsylvania anthracite coal (steaming)....	50.4	50.9	51.4	51.9
Low-grade bitum nous coal.....	51.9	52.2	52.4	52.7
Fuel oil (15-deg. A. P. I.).....	53.1	53.2	53.3	53.4
Natural gas.....	55.7	55.4	55.1	54.7
Blast-furnace gas.....	49.0	49.4	49.8	50.3

From this table it is apparent that the value of R_c does not generally vary more than about 8 per cent from R_a . Consequently, since the uncertainty involved in estimating the *average* stack temperature is more than this, R_c may justly be taken the same as R_a for chimney designs and oftentimes also for chimneys in operation; and then at **sea level** where $p_a = 14.7$, the estimated static draft becomes

$$\Delta p' = 7.64 Z_c \left(\frac{1}{T_a} - \frac{1}{T_c} \right). \quad . \quad . \quad . \quad . \quad (624)$$

By using Eq. (624) for chimneys at *high altitudes*, however, a large error will be introduced, as may be easily calculated from Eq. (623) for values of p_a appreciably less than 14.7.

Observe that all these equations for static draft involve only one physical dimension of the chimney—its *height*; and that the *static draft* is the *maximum* one that could be produced by any chimney of a given height when filled with stationary gases of a given density and surrounded by a still atmosphere of a given density.

(c) When a flow of gases takes place through a natural-draft system, each portion of the passageway offers a definite resistance to this flow. The chimney draft must then be sufficient to overcome each of these resistances and, in addition, to accelerate the gases from zero velocity to that at which they are discharged from the stack. In this case, however, conditions are quite different from those existing before flow was established, as may be well illustrated by reference to Fig. 868.

This figure shows the path of the air and gases from the boiler room, through the setting of a conventional water-tube boiler, past the dampers, and through the breeching into the base of the chimney. Certain major obstructions to flow are encountered by the stream of air and gases between the various numbered points on the figure; these are (a) the grate and fuel bed between points 1 and 2, (b) the boiler and superheater between points 2 and 3, (c) the dampers between points 3 and 4, (d) the breeching between points 4 and 5, and (e) the right-angle bend into the chimney between points 5 and 6. Each of these obstructions is responsible for a certain drop in pressure; and the table accompanying Fig. 868 gives a list of the *absolute pressures* and of the *drafts* which may reasonably be expected to exist during normal operation at the various points for a natural-draft system of the form illustrated. The pressure drop or the "draft loss" between any two successive points may easily be found by subtraction. It should be noted that the resistances given here are only the major ones present in the simplest form of modern setting; the addition of other equipment, such as an economizer or air heater, or a change in the location or design of the equipment illustrated, might change the values of the absolute pressure and of the draft considerably and might even necessitate the use of supplementary draft apparatus.

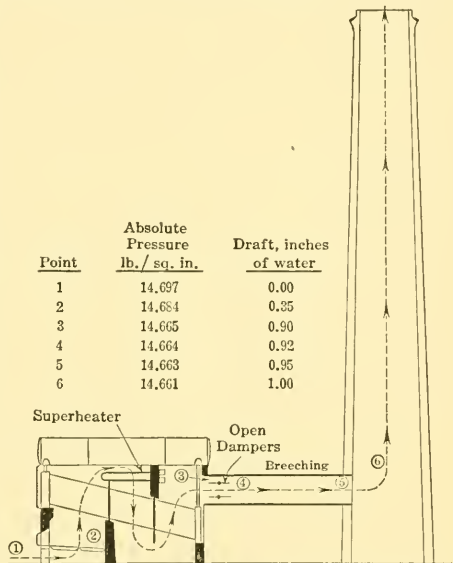


FIG. 868.—Illustration of a Natural-Draft System.

For the case illustrated in Fig. 868 the draft at point 6 must be 1.00

in. of water in order that all the resistances to flow may be overcome. If to this quantity be added the draft required to accelerate the gases and that necessary to overcome the resistance within the chimney itself, the sum will be the *total draft* which must be produced by the chimney in order that proper operation may result. This sum, of course, must not be greater than the static draft which the chimney would be capable of producing at point 2 if the flow through the grate and fuel bed were completely shut off and there were no leakage through the setting. The basic procedure for finding the height of the chimney consists, therefore, in estimating the sum mentioned above and in making the chimney high enough to produce a static draft at least equal to this *total draft* as shown by the following equation for this case, all the terms of which are expressed in inches of water:

$$\Delta p' = D_F + D_B + D_D + D_E + D_T + D_V + D_C \quad . \quad (625)$$

in which

$\Delta p'$ = the total draft to be produced by the chimney.

D_F = the draft loss through the grate and fuel bed.

D_B = the draft loss through the boiler and superheater.

D_D = the draft loss through the dampers.

D_E = the draft loss through the breeching.

D_T = the draft loss due to the right-angle bend between points 5 and 6.

D_V = the draft required to accelerate the gases.

D_C = the draft loss due to the frictional resistance within the chimney.

To make Eq. (625) applicable to the *general case* for a natural-draft system it is only necessary to include as many draft-loss terms in the equation as are required to account for all the resistances encountered by the gas stream in the passageway. For any given installation the nature of these resistances is known and the draft losses produced by them may be evaluated.

(d) The **magnitude of the draft lost** through various parts of a *modern steam-generating unit* is given in Table XLVII (page 455), and also in Table LXVI (page 690). In both cases note that this loss is almost directly proportional to the square of the velocity of the gases, as has already been shown by Eqs. (609) and (610), page 804. Additional draft data are also given in Tables LXIII and LXIV, pages 686 and 687.

The draft lost through the **fuel bed** depends upon the kind of fuel used and upon the rate of firing, as shown by the curves in Fig. 561 (page 477).

The draft losses through the **common types of boilers and flues** are given in most handbooks on mechanical engineering and may also be estimated by the methods given in the preceding chapter. The probable losses through **unusual types of boilers, economizers, air heaters**, and other apparatus to be used in the draft system can be obtained from experienced manufacturers who supply such equipment.

(e) The **pressure drop due to friction in a chimney** forms one of the losses in the draft system, and this may best be studied by means of an example which will show both the method of calculating such drops and the magnitude of this particular drop. Consider that the chimney is of brick and 12 ft. in diameter, and that the gases flowing through it have an average velocity of 25 ft. per sec., an average temperature of 500 deg. fahr., and an average absolute pressure of 14.5 lb. per sq. in.

The solution first requires the determination of the density (ρ) and the viscosity (μ) of the gas. Assuming the composition of the gas to be such that the gas constant (R_c) is 52, then $\rho = P / R_c T = (144 \times 14.5) \div (52 \times 960) = 0.042$ lb. per cu. ft. Since nitrogen is the chief constituent of the chimney gases, the viscosity of these gases may be taken to be the same as nitrogen; hence, from Fig. 832 for a temperature of 500°, $\mu = 18 \times 10^{-6}$ poundal sec. per sq. ft. Therefore,

$$\text{Reynolds' number} = \frac{Dv\rho}{\mu} = \frac{12 \times 25 \times 0.042}{18 \times 10^{-6}} = 7 \times 10^5.$$

Then, for a brick chimney of the given diameter, the friction factor is found from curve 4 of Fig. 847 to be $f = 0.016$. Consequently the friction drop in pressure per 100 ft. of chimney becomes, from Eq. (608) (page 804),

$$\Delta P = f \left(\frac{L}{D} \right) \frac{\rho v^2}{2g} = 0.016 \times \frac{100 \times 0.0416 \times (25)^2}{12 \times 2 \times 32.17} = 0.054 \text{ lb. per sq. ft.,}$$

or in the usual unit, $\Delta p' = 0.0104$ in. of water at 80 deg. fahr. This constitutes a reasonable drop through a chimney of this type.

513. The Measurement of Draft.—(a) The **slight differences in pressure** which exist in a draft system, either between the confined gas and the outer air or between the gas streams at two sections of a passageway, make possible the measurement of the draft by means of the difference in level of two columns of a liquid such as kerosene, oil, or water, in some form of a U-tube, the legs of the U-tube being connected respectively to the two points between which the pressure difference is to be determined. The use of this apparatus involves a knowledge of the specific gravity of the liquid in the U-tube, in order that the draft may be expressed in terms of the usual unit, **the inch of water**, which is a very satisfactory one for expressing small differences in pressure, even though water itself may not be a satisfactory fluid to use in some types of draft gages.

(b) The **pressure equivalent of an inch of water** naturally depends upon its temperature and is as follows:

Temperature, t in deg. fahr.....	40	60	80	100	120
Density, lb. per cu. ft.....	62.43	62.38	62.23	62.00	61.73
Press. equiv. of 1 in. H_2O :					
in lb. per sq. ft.....	5.203	5.198	5.186	5.167	5.144
in lb. per sq. in.....	0.03612	0.0361	0.03601	0.03588	0.03572

Usually the temperature of the water in a U-tube is close to 80 deg. in a boiler room, so that the constant *0.036* from the above table is often a convenient and suitable one to use in such cases.

The pressure expressed in terms of inches of water may also be needed in terms of inches of mercury, and vice versa. The conversion factor is usually desired in terms of the mercury at a temperature of 32 deg.; but whether at this temperature or at that of the water, the factor may be obtained from the following table:

Temperature, t in deg. fahr....	32	40	60	80	100	120
Press. equiv. of 1 in. of Hg, in lb. per sq. in.....	0.4912	0.4907	0.4898	0.4888	0.4878	0.4868
Inches of H_2O , equiv. to 1 in. of Hg both at t	13.595	13.581	13.568	13.574	13.595	13.628
Inches of H_2O at t , equiv. to 1 in. of Hg at 32 deg. fahr....	13.595	13.595	13.606	13.640	13.690	13.751

(c) The **use of U-tubes**, commonly called **manometers**, to measure very small differences in pressure of a gas flowing in a closed passage, such as a boiler setting or chimney, requires special care in their installation and in interpreting their readings. The connection to the gas passage should be by means of a nipple having its inner end set flush with the inside wall and its axis at 90 deg. from the direction of the flow of the gases. It is important that the inner edge of the nipple be free from burrs. Some authorities specify a nipple having a sharp edge; others prefer one having rounded corners. All connections from this nipple to the gages should be free from leakage. However, small leaks are difficult to detect, and cause a greater velocity and consequently a greater loss of pressure in a small tube than in a large one; hence many permanent installations of draft gages are made with relatively large pipes, say $\frac{1}{2}$ in., instead of small tubing. Special attention also needs to be given to the heights and densities of the various gas

columns involved, when a very small drop in pressure between any two points in a gas passage is being measured. The simplest case is that of horizontal flow which will therefore be considered first, after which the case involving change in elevation will be discussed.

(d) For the **horizontal flow** of a gas in a closed passage, such as that indicated in Fig. S69, assume that the pressures P_1 and P_2 , at points 1 and 2, respectively, are to be measured by means of different systems of manometers in which the fluids used may, in general, be designated as *gas X* and *liquid Y* having densities ρ_x and ρ_y , respectively. Various kinds and combinations of manometers may be employed, as shown by the figure, the preferred selection in any application depending upon circumstances; either the two open manometers or the single differential one would be sufficient to determine $P_1 - P_2$.

Since the liquid *Y* is used in the gage it is the one employed to express pressures, and 1 in. of gas *X* will, therefore, represent a pressure of ρ_x/ρ_y inch of liquid *Y*.

Assuming no leakage in any of the manometer connections, and that the temperature of the gas *X* is the same in both leads to the differential manometer, the

difference in absolute pressures, as measured by L_D inches of liquid *Y* in the differential manometer, is

$$p'_1 - p'_2 = \left(1 - \frac{\rho_x}{\rho_y}\right)L_D. \quad . \quad . \quad . \quad . \quad (626)$$

This same result may also be obtained by means of the two U-tubes that are open to the same barometric pressure ($B_1 = B_2$). Thus, in inches of fluid *Y*,

$$p'_1 - p'_2 = \left(1 - \frac{\rho_x}{\rho_y}\right)(L_2 - L_1); \quad . \quad . \quad . \quad . \quad (627)$$

hence, $L_2 - L_1 = L_D$.

In Eqs. (626) and (627) the magnitude of the term $\left(1 - \frac{\rho_x}{\rho_y}\right)$ will be about 0.988 when fluid *Y* is water and fluid *X* is air at ordinary atmospheric pressure and temperature. For such cases this term is therefore commonly taken as unity. On the other hand, if fluid *X* be water and fluid *Y* be mercury, both at 80 deg. fahr., then $\left(1 - \frac{\rho_x}{\rho_y}\right)$

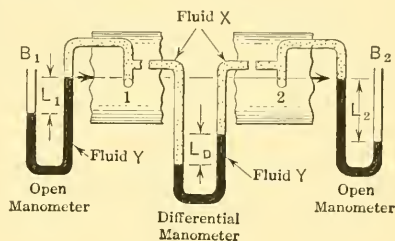


FIG. S69.

becomes 0.927, and thus an error of more than 7 per cent is involved by considering this term unity.

(e) When **different elevations** are involved between the two points

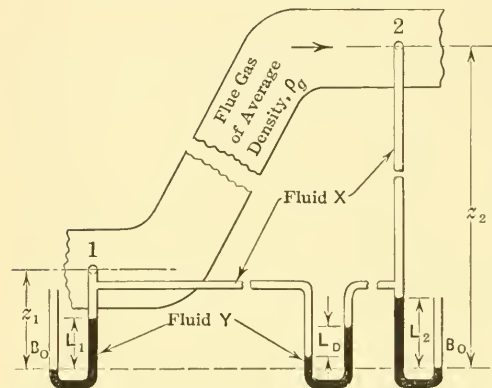


FIG. 870.

for which the difference is gas pressure is desired, the general case may be represented diagrammatically as in Fig. 870. Assume that the points 1 and 2 are located at the respective distances z_1 and z_2 feet above some convenient level, such as that of the instrument board in the boiler room, where the barometer reads B_0 inches of mercury. If the density of mercury in the barometer

is C times that of fluid Y , and the temperature is the same in all the leads to the manometers, the absolute pressures at 1 and 2, in inches of liquid Y , are:

$$p'_1 = CB_0 - \left(1 - \frac{\rho_X}{\rho_Y}\right)L_1 - 12 z_1 \frac{\rho_X}{\rho_Y}, \quad (628)$$

and

$$p'_2 = CB_0 - \left(1 - \frac{\rho_X}{\rho_Y}\right)L_2 - 12 z_2 \frac{\rho_X}{\rho_Y}. \quad (629)$$

Hence

$$p'_1 - p'_2 = \left(1 - \frac{\rho_X}{\rho_Y}\right)(L_2 - L_1) + 12(z_2 - z_1) \frac{\rho_X}{\rho_Y}. \quad (630a)$$

Or, from the differential manometer,

$$p'_1 - p'_2 = \left(1 - \frac{\rho_X}{\rho_Y}\right)L_D + 12(z_2 - z_1) \frac{\rho_X}{\rho_Y}. \quad (630b)$$

From these equations it follows that again $L_D = L_2 - L_1$; but note that for this case L_D is *not* the real difference in the absolute pressures, $p'_1 - p'_2$, because, in general, the flue gas density ρ_g is not the same as that of the gas X in the manometer leads, and z_1 is not equal to z_2 . If the temperature is not the same in the two leads, correction for

the difference is necessary if great accuracy is important. Furthermore, any difference in the gas velocities at 1 and 2 will also affect the difference in static pressures, as shown in the following section, and correction for it is usually required.

514. Velocity, Static, and Total Pressures.—(a) When any fluid fills a closed conduit but no flow takes place, a pressure gage may be attached by a tube at any angle to this conduit, and the resultant reading of this gage will be the **static gage pressure** if the gage be correct; but when a flow takes place there will be a *velocity component* that affects the gage reading *unless* the connection is at right angles to the direction of flow, as shown at (b) in Fig. 871. By connecting the gage so that the fluid tends to flow directly into an open end of the impact tube,³ as at (a), the monometer indicates the sum of the static and velocity pressures; this sum is very logically called the **total pressure**.⁴ If the gage is connected as at (c), so as to indicate the difference between the total and static pressures, the result will be the measured **velocity pressure**.

(b) The calculation of the velocity pressure is based on the definition that the velocity pressure of any fluid is the increase in pressure that would result if the average velocity in the direction of flow could be gradually reduced to zero without turbulence or

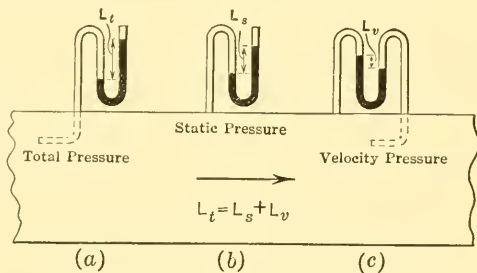


FIG. 871.

heat transfer. If these conditions hold for the apparatus shown at (c) in Fig. 871, the pressure in the impact leg then exceeds that in the static leg by an amount that represents the *complete* conversion of the specific velocity energy into the internal energy (*i*) and into the energy represented by the product of pressure and specific volume ($P\bar{V}$). Very often this increase in pressure is extremely small, hence, for such cases the heat transfer and the change in internal energy, density, and temperature may be considered negligible, because they are smaller than the errors of measurement. For such cases, the velocity energy is very closely represented by an increment in the product of the pres-

³ For a discussion of the advantages and applications of the impact tube, see "The Impact Tube," by Sanford A. Moss, Trans. A.S.M.E., 1916, pp. 761-797.

⁴ This sum is sometimes designated as the "impact pressure," and also as the "dynamic pressure"; but the term "total pressure" is becoming more generally used because it cannot cause any confusion.

shown in Fig. 872. This model is designed to be mounted on a gage board; and since the accuracy of the indications depends upon the careful adjustment of the instrument on the board, provision is made for the leveling of the frame before it is permanently fixed. The inclined tube, which is open to the atmosphere at one end, is attached at its lower end to a vertical reservoir of large diameter as compared with that of the tube. When the space above the liquid in the reservoir is connected to the gas passageway in which a pressure less than atmospheric exists, the level of the liquid in the reservoir rises while that in the inclined tube falls. If the tube had an inclination of 90 deg. to the horizontal, the relative change in level in the reservoir and the tube would be inversely proportional to the cross-sectional areas of the two containers, since the increase in the volume of liquid in the reservoir must equal the decrease in the volume of liquid in the tube; however, since the tube has only a small inclination relative to the horizontal, a given *vertical* drop in the level of liquid in the tube results in a much greater *axial* displacement of this liquid level; hence the relative change in level in the reservoir and the tube is *multiplied* considerably, and a scale placed parallel to the axis of the tube may be so arranged that a pressure difference amounting to 0.01 in. of water can be read directly, while a difference of 0.005 in. may readily be estimated.

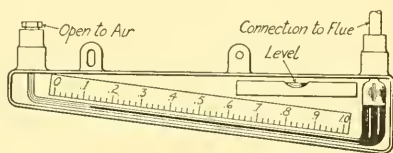


FIG. 872.—Inclined-Tube Draft Gage.

515. Mechanical Draft Fans.—(a) Either forced-draft or induced-draft fans, or both, are now universally used in large intensively operated steam plants. These fans may be driven by steam turbines, steam engines, or electric motors, the choice of drive depending largely upon the type, size, and speed of the fan, and upon considerations of the energy balance of the plant. The two general **types of fans** used are the *centrifugal* and the *propeller*.

(b) The **centrifugal fan** is the type used almost exclusively in central stations for producing both induced and forced draft. Such a fan consists primarily of a bladed wheel which revolves in a spiral casing having a peripheral discharge, and an axial inlet on one side or on both sides. These fans are of two general designs, which are known, respectively, as *steel-plate fans* and *multi-vane* or *multi-blade fans*. In **steel-plate fans**, which resemble paddle wheels (see Fig. 873), the fans proper, or rotors, are of comparatively large diameter; have but a few blades (5 to 12) which are relatively long, radially; and rotate at lower speeds

than do fans of equal capacities of the other type. **Multi-blade fans** (see Fig. 874) have from 20 to 64 blades which are quite narrow radially and long axially; they are of smaller diameter than steel-plate fans of the same capacity and are designed for high-speed drive. Slow-speed fans may be driven by direct-coupled steam engines, by belt from electric motors, or by steam turbines with reducing gears. High-speed fans may be driven directly by motors or steam turbines; and for any type of fan the drive may be either at constant speed or at variable speed, as conditions require, and with either local or remote control.

Forced-draft and induced-draft fans with double inlets are shown in Figs. 875 (a) and (b), respectively, the induced-draft fan having rectangular inlet boxes. The casings can be arranged to discharge at any desired angle. The gases handled by an **induced-draft fan** have a

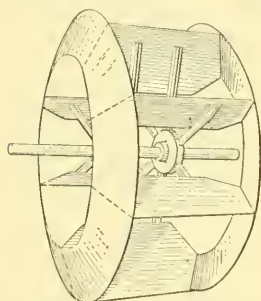
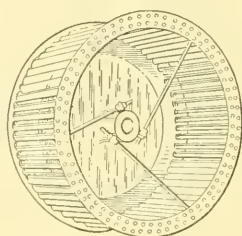
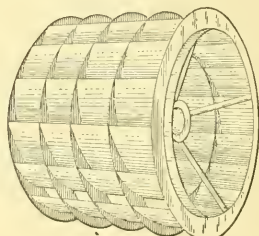


FIG. 873.—Steel-Plate Fans.



(a)



(b)

FIG. 874.—Multi-Blade Fans.

(a) "Sirocco," Single Inlet.

(b) Double-Inlet Fan, Backward-Curved Blades.

much larger volume and a slightly greater weight than those for the combustion air introduced into the furnace; hence these fans must be larger in size than forced-draft fans serving the same unit; furthermore, they must be specially constructed so that their bearings will be protected from the hot and dust-laden flue gases handled. Fans of the steel-plate type are often preferred for induced-draft work because the wide passages between their blades do not easily become clogged with soot and ash dust which reduce capacity and increase power consumption; furthermore, erosion of the blades may be easily avoided through the application of protective plates. **Forced-draft fans** are commonly of the high-speed multi-vane type with full backward-curved blades, since this type has very favorable operating characteristics, as will be explained in detail later.

(c) **Propeller-type** fans of small size are often employed as under-grate blowers, and in such applications are commonly driven by either

direct-coupled turbines or motors. They are used primarily to supplement natural draft during emergency or peak-load operation.

(d) The **horsepower** of a fan, when not otherwise specified, generally means the **shaft horsepower**, which is the power supplied to drive the fan, as contrasted with the **air horsepower**, which is the useful power delivered to the air. Since the rise in pressure in a fan is only a few inches of water, the corresponding reduction in the volume of the air is extremely small; hence if this change in volume be considered negligible, the *air horsepower* becomes

$$\text{a.hp.} = \Delta P V \div 33,000, \quad . \quad . \quad . \quad . \quad . \quad (633)$$

where V = the volume of the air entering the fan, in cu. ft. per min.,
and ΔP = the rise in pressure produced by the fan in lb. per sq. ft.

If this value of ΔP is the rise in the *total pressure*, then Eq. (633) gives the **total air horsepower**; and if ΔP represents the rise in *static pressure*,

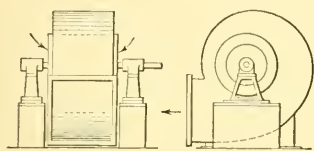


FIG. 875(a).—Forced-Draft Fan.

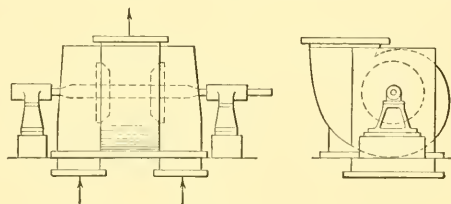


FIG. 875(b).—Induced-Draft Fan.

then this equation gives the **static air horsepower**. If the rise in pressure in Eq. (633) be given as $\Delta p'$ inches of water at 70 deg. fahr., then the air horsepower becomes

$$\text{a.hp.} = \Delta p' V \div 6356. \quad . \quad . \quad . \quad . \quad . \quad (634)$$

If the fan takes the air directly from the atmosphere, as indicated in Fig. 876, the initial velocity pressure is zero, and consequently the value of $\Delta p'$ might then be obtained from one impact tube located as shown. The fan is thus properly credited with the velocity pressure that it has produced in the delivery line. On the other hand, if the fan receives the air from an inlet duct, as in Fig. 877, the rise in the total pressure $\Delta p'$ could then be obtained by a differential manometer connected to impact tubes on the inlet and delivery ducts as shown.

The *shaft horsepower* of a fan is measured by some form of dynamometer, and is greater than the total air horsepower by the amount used

in producing turbulence and in overcoming the fluid friction of the air and the mechanical friction of the fan.

(e) The **efficiency of a fan** is the ratio of its air horsepower to its shaft horsepower; hence the **total efficiency** is

$$e_t = \frac{\text{total air horsepower}}{\text{shaft horsepower}}, \quad (635)$$

and the **static efficiency** is

$$e_s = \frac{\text{static horsepower}}{\text{shaft horsepower}}. \quad (636)$$

The maximum values of the total efficiencies range from about 40 to 80 per cent for various types and sizes of fans.

(f) **Size** and **capacity** are two important factors entering into the selection of centrifugal fans for given kinds of service; and although there are several sizes of any given type of fan which will handle the required quantity of air per minute at the specified discharge pressure,

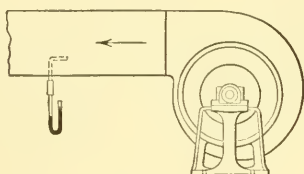


FIG. S76.

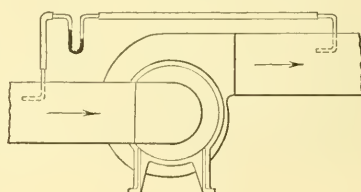


FIG. S77.

one particular size will have the most favorable combination of rotative speed, shaft horsepower, and cost, at the same time meeting the other requirements of the installation.

The **size** of a fan is specified by giving the diameter of the wheel in inches; and the efficiencies of the same type of fan are very nearly independent of the size for diameters ranging from 36 to 120 in. For any specified size and type there is some particular rate of air delivery which gives the maximum efficiency, as is shown in the following section.

The **capacity** of a fan is usually expressed in terms of the number of cubic feet of air which may be discharged per minute when the inlet pressure and temperature have certain specified values. The inlet pressure is taken as one standard atmosphere and the inlet temperature is commonly taken as 70 deg. fahr. Sometimes the results of fan tests or the values in the capacity tables issued by fan manufacturers are based upon a standard condition of the air which differs from that just given; hence a careful study should be made of such data before

they are used, in order that the basis upon which the values were determined may be clearly understood. There is an increasing tendency toward the use of 68 deg. fahr. as the standard temperature for determinations involving apparatus such as centrifugal fans, since this figure happens to correspond exactly to 20 deg. cent. which is taken as a standard in several European countries.

The **rated capacity** of a fan is the number of cubic feet of air it will handle per minute at the standard condition, when it is running at its rated speed and is discharging the air at the rated static pressure. The point of rated capacity is usually taken at a discharge slightly greater than that corresponding to the point of maximum efficiency.

516. Characteristics of Centrifugal Fans.—(a) The **curvature of the blades** determines, to a large extent, the field of usefulness of a centrifugal fan. The blades which are in common use are (1) the straight radial blade, (2) the forward-curved blade, (3) the curved

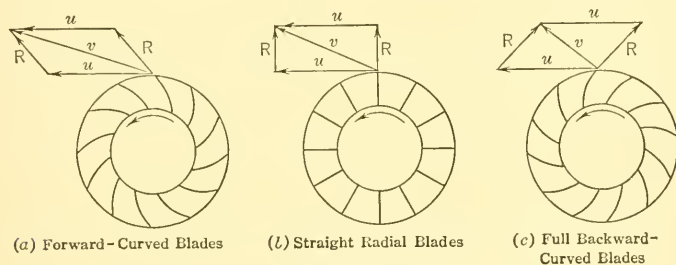


FIG. 878.

blade with radial tip, (4) the partially backward-curved blade, (5) the full backward-curved blade, and (6) the blade with double curvature. Shapes 1, 2, and 5 may be taken as representing the three distinctive types for the purpose of analyzing the action of the air which is flowing through the fan.

(b) The **velocity relationships** produced by blades 1, 2, and 5 are shown in Fig. 878. Here the vector diagrams connecting the relative air velocity (R), the blade velocity (u), and the absolute air velocity (v) indicate that for both the forward-curved blade and the straight radial blade the absolute air velocity is greater than the blade velocity, while for the full backward-curved blade it is considerably less. The excess of velocity (v) over velocity (u) is greatest for the forward-curved blade, and diminishes as the blade is gradually flattened out and bent backward, until velocity (v) finally becomes smaller than velocity (u). This change in the velocity relationships with a progressive change of blade curvature has a definite influence upon the static pressure which

can be produced at the blade tip for a fan operating at a given peripheral speed. In all cases the value of (v) is greater than the velocity of delivery from the fan outlet, hence some of the velocity pressure of the air leaving the blade tips is converted into static pressure within the fan casing. This conversion occurs with considerable turbulence, and hence rather inefficiently; it is least with fans having full backward-curved blades, because for this type (v) is smaller than (u) ; hence such fans have higher static efficiencies than the others, but they must be operated at higher peripheral speeds to produce the same static pressure. However, the static efficiencies of the other fans can be improved by using specially shaped divergent discharge pipes. Despite their somewhat lower static efficiencies, these other fans have several fields of application either because of their relatively low rotative speed, for a given pressure and capacity, or because of other characteristics which may make their selection desirable. Thus the inherent velocity relationships existing for each kind of blade curvature have a marked influence upon fan performance, and hence make one type more suitable than another for a given class of service, as will be discussed in detail later.

A fan should always be designed to minimize the turbulence and fluid friction within it, yet at the same time the design has to be such that the fan is not too expensive to build. The greater the turbulence and friction in any particular design, the greater will be the power required to drive such a fan for any given rate of delivery and increase in pressure.

(c) The **operating characteristics** of centrifugal fans are shown by the relations existing between the following quantities: (1) the volume of air delivered, (2) the static pressure produced, (3) the total pressure produced, (4) the horsepower required, and (5) the rotative speed. These characteristics are influenced to a considerable extent, however, by numerous factors which are beyond the control or calculation of the fan designer; hence the exact values of volume, pressure, power, and speed obtainable during operation can be determined only by tests of the various types of fans; and the relationships between these quantities are best shown graphically.

(d) **Characteristic curves** plotted from the data found during fan tests form the basis for the proper choice and operation of centrifugal fans. The form of these graphs is influenced mainly by the curvature of the blades and by the shape of the inlet and outlet ducts; however, the blade-curvature exerts the predominating influence on the graphs and thus limits the kind of service for which a given type of fan is best suited.

Typical sets of characteristic curves for **constant-speed operation** of fans will be shown later (in Figs. 879, 880, and 881). In one method of presenting these curves (Fig. 881) the pressures, horsepower, and efficiency are plotted as percentages of the values obtained at the rated capacity of the fan; another method of recording fan test data is to plot these quantities in percentages as functions of the percentage of wide-open volume, as in Figs. 879 and 880. Characteristic curves can also be plotted with the actual values of the various measured quantities as coordinates, but when plotted on the percentage basis, a single set of curves expresses the performance of all the fans of a given type, regardless of size and capacity. Thus, having the pressure, horsepower, and efficiency at the rated, or other capacity (e.g., as obtained from a manufacturer's tables or some other source), the percentage curves can be used to determine the values of the desired quantities at any other rate of delivery.

(e) The **selection of a fan** for a given type of service is based chiefly upon a study of the characteristic curves, although several other important factors also enter into the problem, such as the possession of the proper strength and mechanical reliability without excessive first cost, and the ability to deliver the required volume of air or gases at the proper pressure, with high efficiency. Additional requirements for successful operation depend upon the kind of service involved and may include the following: (1) ability to run in parallel with other fans while feeding into a common duct, (2) a reserve of pressure to take care of increasing resistance in the system, (3) the ability to be driven directly by a high-speed steam turbine or electric motor, and (4) the ability to furnish an increasing quantity of air without overloading the driver.

(f) The **requirements of a forced-draft system** are such that a fan to serve it must in most cases satisfy all four of the special requirements given above. This makes the best choice the type having full backward-curved blades; and the characteristic curves (Fig. 879) of this type of centrifugal fan will show clearly the reason for this choice. Since the pressure curves rise when the volume delivered is reduced, until a very low rate of discharge is reached (say 40 per cent), the fan can, without changing speed, overcome an increased resistance such as that offered by a badly caked fuel bed on an underfeed stoker. This pressure characteristic also causes such a fan to deliver its proper share of the air supply at all times while running in parallel with other fans and feeding into a common duct, even though its own outlet pressure may vary somewhat from that of the duct. Furthermore, the horsepower curve of this type shows that as the delivered volume increases the power

$$\begin{aligned}
 WHP &= K W D \\
 dWHP &= K D dW + K W dD \\
 \frac{dWHP}{dW} &= K D + K W = K(D+W) = 0 \\
 &\quad D=W
 \end{aligned}$$

required rises to a maximum and then falls off; thus it is impossible to overload the driver with a fan of this kind. This type shows a high efficiency and, for a given static pressure, has the highest peripheral speed of any type; hence it lends itself very well to high-speed operation and to the use of direct drives.

All the other types of fans previously listed have continuously rising horsepower characteristics and therefore do not possess the "self-limiting" feature which is shown by the fan having full backward-curved blades; thus any other type of fan may, if improperly handled, overload its driver. For example, if the pressure in the system falls suddenly on account of the formation of holes in the fuel bed, these

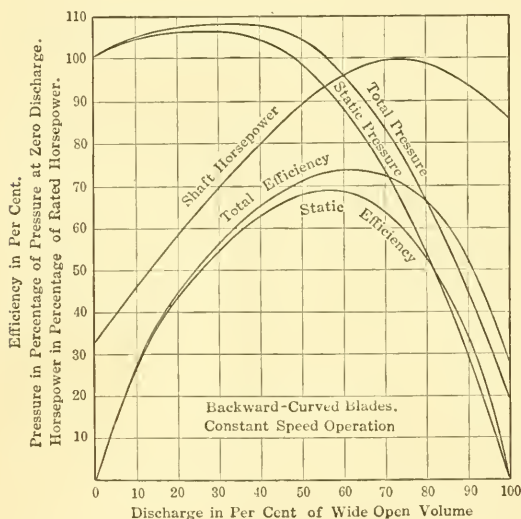


FIG. S79.

other fans will begin to furnish a larger quantity of air than is required for combustion and the horsepower will increase rapidly with consequent overloading of the driver; and this occurs, of course, with the speed unchanged.

(g) The fan having **forward-curved blades** possesses an inherent instability just below the point of maximum efficiency, as shown by the dip in its pressure curve in this region (see Fig. 880). This character-

istic is due to the close relationship which exists, for this type of fan, between the volume of air delivered and the absolute velocity of the air leaving the blades, and may be explained as follows: With a fan running at constant speed, a reduction in the rate of discharge causes a considerable reduction in the absolute air velocity and, hence, a considerable lowering of the static pressure at the fan outlet, because there is less velocity pressure to be transformed into static pressure in the fan casing. The total pressure in the fan casing may thus fall below the total pressure in the draft system; consequently air would flow from the system back into the fan casing. A backward flow would lower the pressure in the system; then the fan would increase its output and would again operate normally until the delivered volume exceeded the quantity of air being

handled by the system, at which time another period of instability would begin. This phenomenon, which is called *surging*, prevents the successful use of fans with forward-curved blades in parallel operation.

Furthermore, this drop in pressure with decreased volume of air delivered means that the fan has no reserve of pressure to overcome increased resistance within the system. However, for induced-draft installations where these fans may be used *singly* and where the resistances to flow are fairly constant, this type is satisfactory. It has the lowest peripheral speed for a given static pressure of any type, since a

large conversion of velocity pressure to static pressure takes place in the casing, and thus it may be operated at a moderate rotative speed,

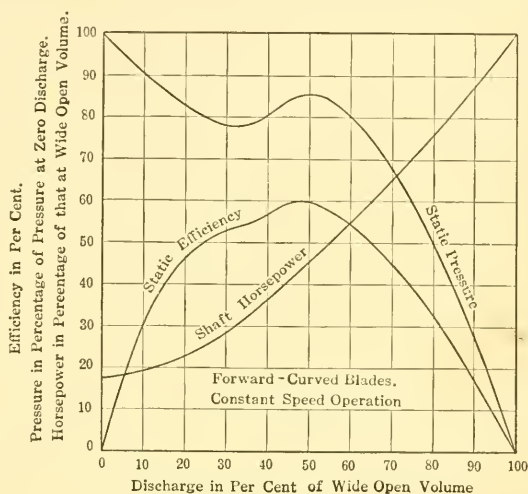


FIG. 880.

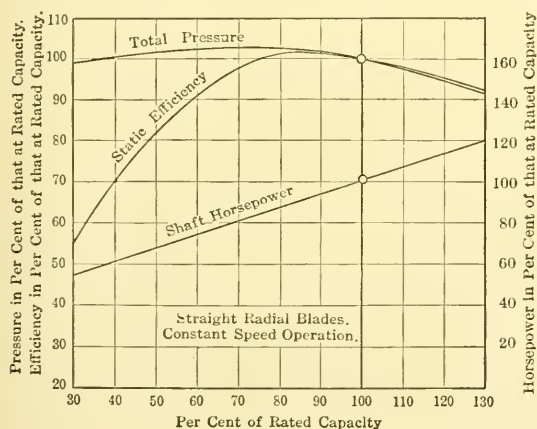


FIG. 881.

characteristics (see Fig. 881) although those of the latter type are influenced to some extent by the amount of curvature given to the blades. Both types exhibit the characteristic of rising pressure with reduced vol-

which is an advantage in induced-draft systems where deposits of soot and cinders within the passages of small high-speed blades tend to reduce the efficiency and to throw the fan wheel out of balance.

(h) Fans with straight radial blades and those with partially backward-curved blades have almost identical performance

ume of air delivered, which fits them for parallel operation; furthermore, each thus has a desirable reserve of pressure. The type with *straight radial blades* is inherently a slow-speed machine although it is now being made smaller and for higher speeds than formerly; the type with *partially backward-curved blades* runs efficiently at a fairly high rotative speed. Neither of these types, however, gives as high an efficiency as does the type having full backward-curved blades. The straight-radial-blade type is a good all-purpose fan and is especially useful for induced-draft work because of the free openings between the blades. Fans having *blades with double curvature* (i.e., blades curving forward at the base and back-

ward at the tip) have characteristics much the same as those with straight radial blades.

(i) The fan having **curved blades with radial tips** gives a *total pressure* which is almost constant over a wide range of capacity, as shown by the flat portion of the curve ⁵ in Fig. 882. Within the region of constant total pressure, a large variation in the volume of air discharged may take place without a corresponding inverse change in the total pressure; hence the

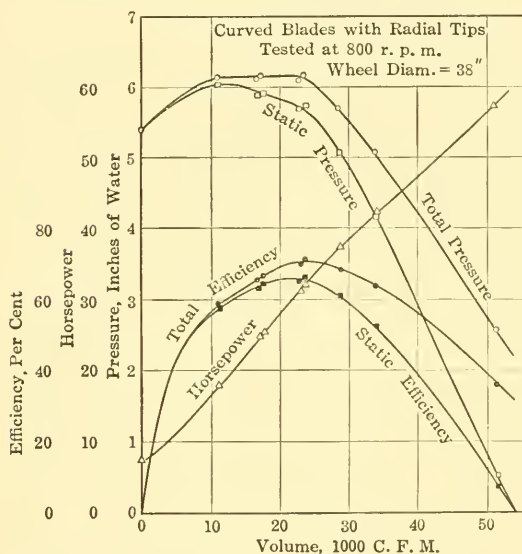


FIG. 882.—Characteristics of a Fan Having a Free Inlet and Curved Blades with Radial Tips.

outlet pressure cannot be relied upon to exert a stabilizing influence upon the rate of discharge, and thus prevent a fan from transferring a part of its normal load to other fans with which it may be operating in parallel. It should be noted, however, that the shape of the inlet box may change the pressure characteristic of this type of fan considerably; hence the disadvantage just discussed may be eliminated and the usefulness of the fan may thus be increased. An illustration of the effect of adding one type of box is shown in Fig. 883.

⁵ See "Influence of Inlet Boxes on the Performance of Induced-Draft Fans," by Marks and Winzenburger, presented at the December meeting of the A.S.M.E., 1931.

However, any inlet box reduces the total efficiency for a given rate of discharge; therefore an improved pressure characteristic may be obtained but at the expense of an increased power input.

The curves in Fig. 883 were obtained when the fan was fitted with a rectangular inlet box having no guide sheets. A study of these two figures will clearly indicate the limitations imposed upon the performance of the fan by the addition of the specified inlet box. This is of special significance in induced-draft installations where the stack gases must be conducted to the fan through some form of closed duct. The purchaser of a fan should know not only the characteristics of the fan with free inlet but also those existing with the specified inlet box attached. In the case given, when the rectangular inlet box was in place the speed had to be increased 12.2 per cent and the power input raised 10.5 per cent to obtain the same performance at any delivered volume as was given by the fan when operating at normal speed with a free inlet.

(j) Under **variable-speed operation** the following relations hold

approximately for a fan of a given size: The capacity varies directly as the rotative speed (N), the pressure produced varies directly as N^2 , and the power required varies directly as N^3 . These relations indicate that, when only a part of the full capacity of a fan is needed, it is desirable to operate the fan at the lowest possible speed at which the required output can be produced. Often, however, the operating conditions are such that the speed cannot be varied or that a variation is not feasible; in such cases constant-speed fans must be used, and the delivery must be regulated by means of dampers. The speed of a fan drive, and the percentage opening of the dampers, can be adjusted either by hand at the fan or by automatic remote control.

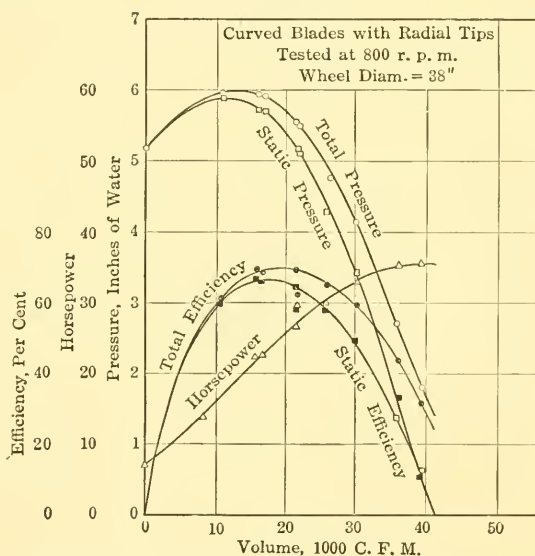


FIG. 883.—Characteristics of a Fan Having Rectangular Inlet Boxes without Guide Sheets.

(k) **Movable inlet vanes** of various types have been recently introduced as a simple means of controlling the volume of gas handled by a fan, and also to control the direction of the gas at entrance. This method of volume control enables the fan to be run at constant speed while requiring less power at low loads than when the volume is controlled by throttling at the outlet. The constant-speed motor also costs far less than the variable-speed one with its control equipment.

517. Propeller Fans.—(a) The **propeller type** of fan has a number of features that make it especially attractive for producing forced draft when the volume of air required is not too great. The very compact form of the fan, especially when driven directly by a motor or turbine, enables it to be installed at relatively low cost in a small space very

close to the furnace, so that often all duct work is eliminated. There are also many other applications where the propeller type of fan may be successfully used; and recent improvements have made its efficiency very satisfactory. This type is now available with capacities up to 60,000 cu. ft. per min. with a maximum static pressure of 10 in. of water.

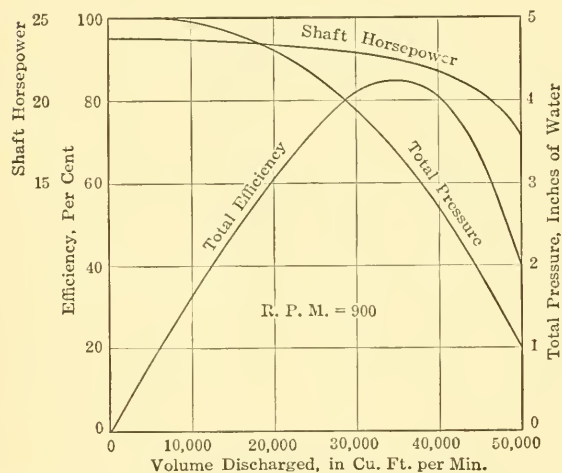


FIG. 884.—Characteristics of a Two-Stage Propeller Fan.

(b) The **efficiency** of the small single-stage type of propeller fan is seldom more than 60 per cent; but by using two stages, and by carefully stream-lining the entrance to and exit from the fan, the efficiency can be materially improved, as is shown by the curves in Fig. 884 for the Aeroto fan⁶ shown in Fig. 885.

518. Locomotive Draft.—(a) The steam locomotive is fitted with equipment for producing an intense **induced draft** in order that high rates of combustion may be obtained, and that large quantities of steam may thus be generated in a boiler of restricted size. This draft is produced in the smokebox (see Sect. 474 (d), p. 674) by means of energy

⁶ These figures were drawn from data given in London Engineering, June 12, 1931, p. 756, in "The Aeroto Forced-Draught Fan."

derived from the exhaust steam as it flows upward through a combination of apparatus which forms an **ejector**.

(b) The **arrangement of apparatus** used to produce the draft is shown in the simple form of smokebox illustrated in Fig. 886. Steam exhausts from each cylinder in turn and blows upward from the **exhaust nozzle** in the form of a jet. The friction between the jet and the stack gases in contact with it enables the jet to lift these gases because of its high velocity; and as the jet continues to ascend it will entrain the gases which it has thus set in motion, finally ejecting them through the **stack extension** and the **stack** to the outer air. At the top of the stack the jet must have enough velocity to propel the mixture of steam and gases to a point in the atmosphere above the stack where diffusion can take place.

In order that satisfactory operation may be obtained, the shape and size of both the exhaust nozzle and the stack extension, and their

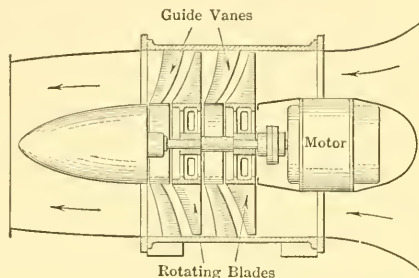


FIG. 885.—The Aero-Two-Stage Propeller Fan.

location with reference to each other, must be such that the ascending column of steam and gases will fill the stack extension and the stack for at least $\frac{2}{3}$ of their combined length, measured from the top down. When this condition exists, a partial vacuum of sufficient magnitude to maintain the required draft is produced in the space between the tip of the exhaust nozzle and the bell of the stack extension. The difference between the absolute pressure at this point and that of the outside air is sufficient to overcome the frictional resistances encountered by the gases flowing through the draft system;

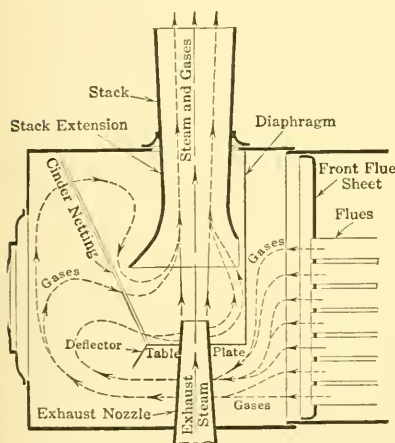


FIG. 886.—One Form of Locomotive Smokebox.

thus the products of combustion move forward from the firebox (see Fig. 720, p. 675), through the flues, to the smokebox, while air for combustion enters the firebox through the grate and fuel bed and through the secondary-air ports.

The **friction losses** within a locomotive draft system may be expressed in terms of the variation in the draft at points between the bottom of the grate and the stack; thus a representative test of a modern freight locomotive ⁷ fitted with a plain exhaust nozzle $6\frac{1}{2}$ in. in diameter and a stack 18 in. in diameter, showed that the draft, in inches of water, was 6.59 at the back of the cinder netting in the smokebox, 6.24 at the front of the netting, 5.39 under the table plate, 3.82 at the back of the diaphragm, and 1.35 in the firebox.

(c) The **back pressure in the main cylinders** is higher than atmospheric because the nozzle which produces the jet of steam in the smokebox constitutes a restriction in the exhaust passages; and this may considerably reduce the useful work done per cycle. If the diameter of the nozzle be increased, other factors being equal, the back pressure is reduced, as is also the draft; and since a certain amount of draft must be maintained in order that the required rate of combustion will result, a compromise must be effected whereby the necessary quantity of fuel may be burned, and, at the same time, the back pressure may be held to as low a value as possible. A large variety of exhaust nozzles have been designed for the purpose of obtaining the most favorable combination of high draft and low back pressure, and some of these designs have proved to be much more effective than others.⁸

It should be noted that the most effective nozzle is one which produces a jet having the largest possible *surface area* for contact with the surrounding gases; in addition the nozzle must *spread* the jet so that the stack will be filled with the mixture, as previously stated. A favorable result may thus be obtained by the use of a special nozzle of large diameter, which causes a low back pressure, whereas the same draft would necessitate a plain nozzle of smaller diameter, which would cause a much higher back pressure.

In many modern locomotives the **maximum draft** amounts to as much as 10 to 12 in. of water when the plant is being operated under high-output conditions. A draft of this magnitude should be produced with a back pressure of 12 to 15 lb. per sq. in. gage, although back pressures twice as high as these may be obtained if the smokebox design is poor or if operating conditions are unfavorable.

519. Chimneys.—(a) A **self-supporting chimney** is essentially a thin-shelled vertical flue which has the combined loadings of a column and of a cantilever beam. Each vertical section is stressed primarily

⁷ See "Improving Draft Efficiency," by G. W. Armstrong, Railway Mechanical Engineer, September, 1930, p. 500.

⁸ See "Drafting Modern Locomotives," by H. W. Coddington, Railway Mechanical Engineer, June, 1918; see also the reference in footnote 7 above.

by the dead weight of the portion of the structure above it, and by the transverse loading due to the lateral wind pressure on that portion; in addition the chimney may be subjected to vibrations such as those due to the wind or to earth tremors. The lateral wind pressure is commonly taken as 25 lb. per sq. ft. of projected area, for purposes of design. Chimneys are made of brick (usually radial brick), reinforced concrete, or steel plates; and a comparison of Figs. 888 to 890, which illustrates stacks of the different types with the same height and internal diameter, will show roughly the relative thickness, weight, and space occupied with each of the various constructions.

(b) **Radial-brick chimneys** (Fig. 888), which have entirely displaced chimneys of ordinary brick in power-plant practice, are constructed of special blocks which have two of their vertical faces cut to fit radii drawn through the center of the chimney. These blocks, which are composed of suitable material, are molded with perforations, and are so laid in the chimney that they overlap each other (Fig. 887), while the mortar, being pressed into the perforations, locks the blocks together. As compared with a chimney of ordinary brick, the radial-brick type has the following advantages: (1) its shell is thinner and weighs less, (2) the foundation may be smaller, (3) the structure is neater and the individual blocks may be more accurately fitted together, and (4) the construction may be more rapidly executed.

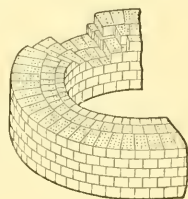


FIG. 887.

The *stability* of such chimneys is dependent almost entirely on their dead weights. The tallest chimney in the world is constructed of radial brick; this structure, which is located at Anaconda, Montana, is 585 ft. high and has a diameter of 60 ft. at the top.

(c) **Reinforced-concrete chimneys** (Fig. 889) have steel reinforcing bars arranged both circumferentially and vertically; the vertical bars are anchored into the foundation which is also of reinforced concrete and is comparatively shallow. As compared with brick chimneys, the reinforced-concrete type (1) is thinner, (2) is lighter, (3) occupies less space, (4) does not require as large a foundation, (5) is free from joints, and (6) may be more rapidly constructed. An inner shell of brick or reinforced concrete is commonly constructed for a portion of the height, but in some cases is entirely omitted. The largest chimney of this type is located in Japan; it is 567 ft. high and 26 ft. 3 in. in diameter at the top.

(d) **Steel stacks**⁹ are either of the self-supporting or guyed types.

⁹ It is quite common practice to restrict the use of the term "stack" to the steel structures and the term "chimney" to the masonry ones.

Self-supporting steel stacks (Fig. 890) are flared at the bottom to a diameter of about one and one-third that of the stack proper and are bolted to their foundations. They are of either riveted or welded construction and are designed to have ample strength even after they have become somewhat corroded. They are usually lined with brick or other material to protect the metal from the heat and the corroding action of the gases. Chimneys of this type are (1) of light weight, (2) easily and rapidly constructed, (3) of relatively low cost, and (4) free from air leakage if properly calked or welded. They must be painted frequently to protect the metal from the weather and from the gases. When ground space is expensive, steam-plant stacks of this kind, and sometimes of the others also, may be mounted on deep girders located at or near the roof level and supported by massive columns between the boilers; in such cases the base of the chimney is often as much as 150 ft. above the ground.

Guyed steel stacks are not usually made larger than 4 ft. in diameter or 75 ft. in height and commonly have four steel guy cables to resist the wind pressure. These stacks are made of relatively thin plates, hence are of low cost, and as they are of light weight and do not depend on foundations for stability, they are often supported directly on the boiler breeching.

(e) The **Thermix stack** (Fig. 891) is the combination, in a single unit, of a mechanical-draft fan and a stack of the *évasé* (vase-shaped) or venturi type, in which the natural draft is augmented by the introduction of a blast either of atmospheric air, as in Fig. 891, or of gas withdrawn from the flue or chimney base; this blast is discharged by a small induced-draft fan into the throat of the stack. Fig. 892 shows a Thermix stack with built-in induced-draft fan and also with provision for the rather effective elimination of cinders, dust, and soot from the chimney gases. Thermix stacks are comparatively short and of light weight; generally they are placed at a level with the top of the boiler, thus saving stack material from the ground to this point and reducing the cost of the breeching. Stacks of this type have been used extensively in Europe, and there have been a few applications in this country. The names *Prat* and *évasé* are both commonly applied to the arrangement shown first.

(f) **Linings** are usually employed in chimneys (except the last two types considered) for at least a portion of their height in order to reduce the heat losses through the chimney walls or to prevent damage by erosion, corrosion, or temperature stresses. If a *masonry chimney* is used which will not resist the corrosive action, or if it is exposed to gas temperatures above, say, 1200 deg. fahr., the lining should extend to the full height of the chimney. As the products of combustion carry

in suspension particles of dust and cinder, the chimney wall opposite the right-angle bend at the exit from the breeching may be rapidly eroded by these particles unless it is protected by a lining for at least a short distance. The lining of a *radial-brick chimney* is usually made

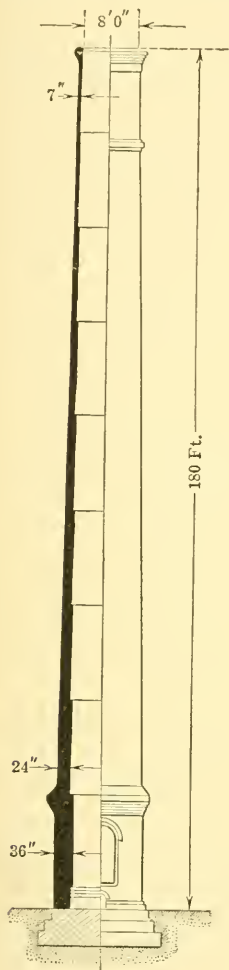


FIG. 888.—Radial-Brick Chimney.

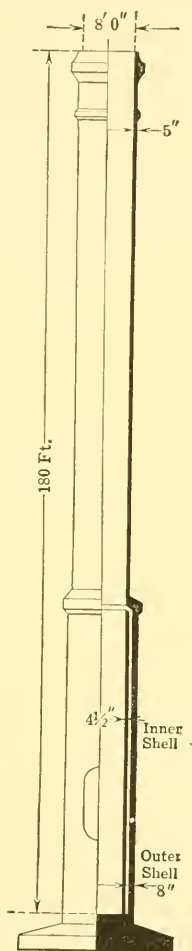


FIG. 889.—Reinforced-Concrete Chimney.

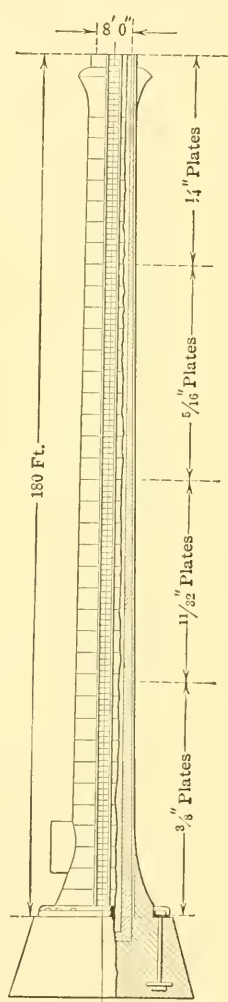


FIG. 890.—Self-Supporting Steel Stack.

also of radial brick, supported on corbels built inward from the chimney shell at vertical intervals of approximately 20 ft., and with an air space of about 2 in. between the lining and the shell. Brick linings in *steel*

stacks may be either self-supporting or constructed in a series of independent bands, each resting on brackets attached to the shell. The brick lining is generally laid in contact with the shell and is grouted thereto. Thinner linings of special materials, such as vitrified asbestos, are also used and are supported on horizontal shelves.

(g) **Chimney proportions** should be such as will meet the requirements of draft and capacity with the least cost for the structure. The diameter needed for a chimney depends on the volume of gases to be handled per unit of time and on the velocity of flow allowed; and the height of the chimney must be sufficient to produce this rate of flow and at the same time maintain the maximum effective draft needed at the flue connection. By using higher velocities, the chimney diam-

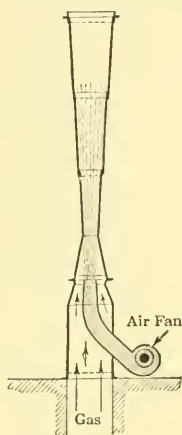


FIG. 891.—Thermix Stack with Air Blast.

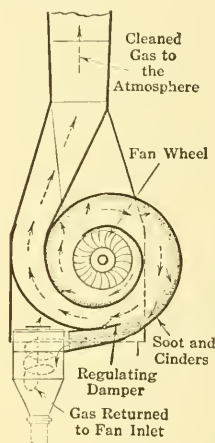


FIG. 892.—Thermix Stack with Cinder Catcher.

eter can be reduced, which would result in a lower cost of chimney if other factors remained the same; but these greater velocities necessitate increasing the height of the chimney, thus entailing an additional expense which partly or wholly offsets that saving. Evidently for each given set of conditions there is some velocity which will give a proportion of height to diameter of the chimney that will entail the least outlay of money for material and labor in building the structure. A satisfactory proportion used with many radial-brick chimneys is to have the height 22 times the diameter. Short chimneys often have a maximum velocity of flow of 20 ft. per sec., and very tall ones 30 ft. per sec., or more. Some authorities consider that the most economical stack is the one meeting the specific requirements as to draft and capacity with the

lowest product of the height (Z_c) and diameter (D) as found by trial. This assumes that the cost of a chimney is proportional to $D \times Z_c$. It has been shown, however, by V. S. Renton,¹⁰ that the cost of radial-brick chimneys is more nearly proportional to $Z_c^2 \times \sqrt{D}$.

(h) **Kent's empirical formula**, formerly widely used, gives roughly the relation between the effective area (A_e), in sq. ft., and the height (Z_c), in ft., of a chimney for a given "boiler horsepower" (HP) rating (see p. 441). This formula is

$$A_e = 0.3HP \div \sqrt{Z_c},$$

and is based on the assumptions that 5 lb. of coal are burned per hr. per "boiler hp.," and that the effective area of the chimney is the actual area reduced by a 2-in. stagnant film of gas adhering to the inner wall surface. Before the formula can be applied, the chimney height must first be determined, as, for instance, on the basis of assumed or known resistances to be overcome. This formula is mentioned in this text merely because it has been widely quoted in the past and because many of the published chimney tables are based on it. It should be used merely as a rough check, if at all.

520. Air Ducts and Gas Breechings.—(a) **Recommended air velocities** in the main ducts supplying stokers are given below. These are in accordance with the figures of the Stoker Manufacturers Association, and are based on an air temperature of 70 deg. fahr., an air pressure of 29.92 in. mercury abs., and a relative humidity of 50 per cent.

Static pressure, in. of water.....	2	3	4	5	6	7
Duct velocity, ft. per min.....	1800	2200	2550	2850	3125	3350

These duct velocities will produce velocity pressures amounting to between 5 and 10 per cent of the corresponding static pressures at the stoker wind boxes; and they should be increased by 1 per cent for each 500 ft. of rise in elevation or for each 10 deg. fahr. increase in temperature. *Branch ducts* leading to stokers should have velocities not exceeding two-thirds of these values. The friction losses through air ducts can be determined roughly by the methods given in the preceding chapter. The walls of air ducts are commonly made of sheet steel or steel plate, but sometimes are of concrete or masonry; the construction should be such that they will always be air tight, and their interior surfaces should be smooth.

(b) **Breechings**, or flues, for the exit gases are usually proportioned to have velocities of flow 20 per cent lower than those in the chimneys themselves. The draft-loss allowances can be computed by the methods already given; often, however, they are approximated by the "rule of thumb" estimates of a loss equal to $\frac{1}{16}$ in. of water per 100 ft. of straight

¹⁰ Thesis: "Chimney Draft," Cornell University, June, 1933.

flue and $\frac{1}{20}$ in. for each right-angle turn. Breechings are commonly made of steel plate reinforced by angle irons or by corrugations; the latter also provide for expansion, but if they are not used, special leak-proof expansion joints must be included. Clean-out or access doors should be provided, and the walls are usually covered externally with 2 in. of block and plastic insulation applied on wire mesh or rod frame and finished with a hard cement surface.

521. Removal of Flue Dust from the Draft System.—(a) The problems of making suitable provision for (1) the deposit of cinders, dust, and soot in the draft system, (2) the removal of these deposits, and (3) the elimination of such matter from the flue gases before they are discharged from the chimney are inherent with intensive furnace operation. These problems are more pronounced with pulverized-coal firing than with stoker firing. The first of them is met by arranging collecting or precipitation pockets in the draft system at points where sudden changes in direction occur; the second is cared for by providing suitable clean-out openings, dust chutes, and hydraulic, compressed air, or suction evacuators; the third requires the use of some kind of flue-dust-removal apparatus, such as cyclone separators, fan-type dust collectors, filters, baffle-plate dry-dust traps, scrubbers, or electrical precipitators.¹¹ Only a very brief description will be given of the various kinds of flue-dust-removal apparatus.

(b) In **cyclone separators** the flue gas is directed tangentially into a circular chamber in which it whirls and from which the clean gas issues centrally at the top, the cinders being discharged from the bottom.¹² These separators are used singly, or in parallel, or in series.

In **vortex collectors** the gases are baffled by a number of vertical plates of special form and arrangement which produce many free spiral vortices, in the low-pressure center of which the dust collects and is then deposited in a settling chamber.

Fan-type dust collectors are induced-draft fans, modified for this special service. In one arrangement, part of the periphery of the scroll is provided with openings which discharge the dust into a collecting pocket; in another arrangement the working faces of the blades have oblique ledges which collect the dust and discharge it laterally into annular collecting grooves in the sides of the casing.

¹¹ For a general review of this subject see "Collecting the Dust from Chimney Gases of Powdered-Fuel Installations," by Toensfeldt, Trans. A.S.M.E., 1928, FSP 50-59, p. 77; also "Smoke and Dust Abatement," by Engle, in Heating, Piping and Air Conditioning, February, 1931, p. 146.

¹² For data on the performance of these separators see "Results of Research on Cyclone Dust Collectors," by L. C. Whiton, Jr., presented to the Metropolitan Section of the A.S.M.E., Feb. 18, 1932; also see Power, March 1, 1932, p. 344.

In **dust filters** the flue gas is passed through metal-mesh or cloth filters which are arranged for easy cleaning.

A **baffle-plate dry-dust trap** consists of a chamber containing a series of vertical stationary sheet-metal elements of special shape, which give abrupt changes in direction to the gases and collect the solid matter therefrom, depositing it in hoppers beneath.

Scrubbers, washers, or wet separators, in their simplest form, consist merely of water sprays located in the stack¹³ or in the flue.¹⁴ One arrangement has multiple baffle plates over which the film of water passes; in another arrangement (the Murray) the gases are projected downward against the surface of a body of water and pass through a narrow passage between a baffle and that surface. A modified arrangement of the latter is known as the Hudson Avenue type of wet cinder catcher.¹⁵

Cottrell electrical precipitators consist of a number of vertical parallel plates of reinforced concrete between which are vertical wires receiving high-voltage direct current. In passing between the plates the dust particles in the gas receive an electrostatic charge which causes them to adhere to the plates, from which they are later precipitated. Although very effective, these collectors are the most expensive of any type.¹⁶

(c) **The selection of types of dust-removal apparatus** was investigated by a committee of the Electricity Commission in England, which surveyed the conditions in that country, Germany, and France, and came to the following conclusions: (1) for pulverized-coal firing the electrostatic precipitation is the most suitable; (2) the use of water sprays for dust collection has certain disadvantages; (3) for stoker firing no dust-removal equipment is necessary if proper conditions of design are met, if the plant is operated with low gas velocities, and if the stack is high; (4) for other conditions the multi-cyclone type of collector is the most suitable.¹⁷ The selection for a given plant depends, however, on many factors influenced by the particular conditions encountered in that plant.

¹³ See *Power Plant Engineering*, Sept. 1, 1930, p. 977, and Sept. 15, 1930, p. 1081.

¹⁴ See *Power*, Nov. 4, 1930, p. 716, and Aug. 1932, p. 396.

¹⁵ "A New Type of Cinder Catcher," *Power*, May 31, 1932, p. 812.

¹⁶ "Electrical Precipitation" (fundamental theory), by A. W. Simon and L. C. Kron, in *Power Plant Engineering*, April 1, 1932, p. 299; "92-98 Per Cent of Dust Removal at Michigan City Plant," in *Power*, March 22, 1932, p. 425; "Harvard Medical Power Extension Precipitates Dust Electrically," by G. K. Sauerwein, *Power*, Aug. 18, 1931, p. 226; "Flue Dust Elimination at Colfax Station," in *Power*, Dec. 30, 1930, p. 1045; "Catching Pulverized-Coal Ash at the Trenton Channel Plant," by H. M. Pier and A. N. Crowder, in *Power*, May 31, 1927, p. 834.

¹⁷ *Engineering* (London), Dec. 16, 1932, p. 714.

CHAPTER XLIV

THE CONDENSER SYSTEM

522. General.—(a) Condensers are used with vapor-actuated prime movers, with steam-jet air ejectors, and with refrigerating, evaporating, and various other kinds of apparatus. The most extensive field of application, however, is with steam turbines, and the discussion in this chapter will be limited chiefly to that field, in which the principal function of the condenser is the reduction of the exhaust pressure, although the recovery of the condensate for use as feedwater for the steam generators is often of almost equal importance. Although any liquid, gas, or vapor that can be obtained cheaply, in sufficient quantities, and at low temperature, can be used as a cooling medium if other considerations permit, water is practically the only material so used in the applications under discussion. The amount of condensing water needed is from 25 to 100 or more lb. per lb. of steam condensed and hence is relatively very large, the requirement of a central station in this respect usually being several times the capacity of the municipal water system of the community served by the station. Condensing steam plants, therefore, especially if large, are almost invariably located immediately adjacent to rivers, lakes, sounds, or other large bodies of water from which an ample supply can be readily obtained and to which the warm water can be returned without affecting the intake temperature. Small inland plants, and even fairly large ones, sometimes use a fixed quantity of condensing water, which is repeatedly recirculated through the condensing system after its temperature has been reduced by means of cooling towers or cooling ponds, such as are discussed in Chapter XLVIII.

(b) The usual condenser serving a steam-driven prime mover is, in effect, merely a dead-end enlargement or chamber at the end of the exhaust pipe, wherein the steam is condensed by cooling water which is either sprayed directly into the steam, or flows through tubes with which the steam comes in intimate contact on their outer surfaces. Thus, condensers are either of the *jet type* (or direct-contact type), in which the condensate mixes with the cooling water and from which both are discharged together, or of the *surface type*, in which the two fluids are kept separate by thin metallic walls and are handled independently.

The condensate recovered from a surface condenser is available for direct use as feedwater for the steam-generating units of the plant. As it is distilled water which is not only quite free from scale-forming impurities but also has been thoroughly de-aerated, it is particularly suitable for this purpose and is especially valuable as such when the steam-generating units are operated intensively at high pressures. Also, utilizing the recoverable heat of the condensate effects a worthwhile saving in fuel consumption. For these and other reasons, the surface type of condenser is the one most commonly used in modern central station practice; never-the-less, special conditions not infrequently justify or require the use of condensers of the jet type.

(c) A **condensing system** includes, besides the condenser itself, the *auxiliary apparatus*, the accessories that directly serve the condenser, and the intake and discharge systems for the condensing water. The auxiliaries include the *condensate pumps*, *condensing-water pumps*, and *air-removal apparatus* that may be part of the installation. The condensing system not only has an important influence on the design and operation of other related apparatus, but also it affects the general layout of the plant; and, as already noted, it usually dominates the selection of the plant site.

523. Measurement of Condenser Pressures.—(a) The **absolute pressure** in a condenser is always the value that is needed when stating condenser pressures accurately or when making condenser calculations; hence it is the preferred term to use when dealing with condensers. Its employment is growing and will undoubtedly eventually eliminate largely the term “vacuum” when expressing condenser pressures. There are two reasons for this change. First, the mere statement of a “vacuum” of so many inches of mercury is incomplete unless the simultaneous barometric pressure is also given or understood to be a definite value; hence this method often becomes indefinite or cumbersome and therefore likely to be inaccurate. In the second place, *accurate mercury gages* that give the absolute pressure directly have been recently introduced, and the results have been found more accurate than those usually obtained by reading vacuum gages and barometers, and making the proper corrections for temperature. Another method of stating the condenser pressure, that is used to some extent in Europe, is to express the vacuum gage reading as a percentage of the standard barometric pressure.¹

¹ Sometimes, though infrequently, the actual vacuum is expressed as a percentage of the maximum vacuum that could be produced with water at the temperature of that entering the condenser and with a *30-in. barometer*, allowance being made for the vapor pressure corresponding to the water temperature. Expressing the vacuum

(b) When **vacuum gage readings** are involved, one should remember that the corresponding absolute pressure is obtained by subtracting this reading from that of a nearby barometer having the same temperature and elevation. When a mercury column L inches long has its temperature increased Δt degrees fahrenheit, the corresponding increase in its length becomes

$$\Delta L = 0.000101(\Delta t)(L). \quad . \quad . \quad . \quad . \quad . \quad (637)$$

This equation enables one to make corrections to the length of a mercury column to find its equivalent length at some other temperature (such as 32 deg. fahr., which is the standard commonly used when no other is specified).

(c) The **standard atmosphere** is defined in the A.S.M.E. Test Code for Condensers as 29.9212 in.² of Hg at 32 deg. fahr., and is equal to a pressure of 14.6963 lb. per sq. in., since the density of mercury at 32 deg. fahr. is 0.4912 lb. per cu. in. If this same pressure is to be equivalent to that of a column of mercury *30 in. high*, the corresponding temperature of the mercury will then be found to be 58.1 deg. fahr. and the corresponding density 0.4899 lb. per cu. in.

(d) The **atmospheric pressure** actually varies widely in any given place from day to day, and sometimes considerable change is noted during a test of a condenser or related equipment. Furthermore, if different altitudes are involved in the comparison of two condensers for which the data are given in terms of the “vacuum,” very erroneous conclusions may be reached regarding the absolute pressures in the two condensers unless the real barometric readings at both places are known. The **relative values** of the barometric readings that may be expected at various elevations above sea level are indicated roughly by the following, in in. Hg:

Altitude, ft.	0	1,000	2,000	3,000	4,000	5,000	6,000	7,000	8,000
Barometer	30.00	28.88	27.78	26.74	25.74	24.78	23.86	22.96	22.10

(e) **Accuracy** in the determinations of the condenser pressures is very essential if careful analyses of condenser and prime mover performances are to be made. With a large turbine a decrease in the back pressure from 2 in. Hg to 1 in. may effect a saving of about 5 per cent in the annual consumption of steam, cooling water, and fuel; thus an error of but 1/10th in. in a barometer reading may seriously affect the accuracy of computations based on such a reading. As the pressure is

percentage with respect to the *actual* barometer reading gives values that are meaningless unless the barometer reading is also stated.

² This corresponds to the International Standard of 760 mm. Hg at 0 deg. cent.

not necessarily the same at all points in the exhaust pipe or in all parts of the condensing chamber, the places of pressure measurement should always be stated and be selected with care.³

524. Condenser Performance.—(a) Under the most favorable conditions, steam alone would be the substance flowing into the condenser from the prime mover; only its latent heat would be removed by the condensing water; this water would have its temperature raised to that of the steam; and the condensate would be used as feedwater without heat loss. Under these conditions, the least amount of heat would be transferred in maintaining the desired exhaust pressure, the condensate would be available at the highest temperature possible at that pressure, and the least amount of cooling water and of energy for pumping it would be used. In actual performance, however, (1) the discharge temperature of the condensing water is always less than that of the exhaust steam by an amount, termed the **terminal temperature difference**, the best value of which depends upon economic considerations; (2) the amount of condensing water used, or *water-to-steam ratio*, is greater than the minimum chiefly because of the smaller rise in the temperature of this water; (3) the temperature of the condensate is less than that of the exhaust steam; and (4) some air and other non-condensable gases enter the condenser with the steam and affect the performance of the apparatus. The first three items will be considered more in detail later.

(b) The **mixture** of air and other non-condensable material entering the condenser with the vapor is commonly called **air** for brevity and it will usually be so termed in this text. Part of this "air" is brought into the system in solution in the feedwater supplied to the steam generator, and part enters through porous castings, leaky glands of steam turbines or stuffing boxes surrounding pistons and valve rods, and through faulty joints of pipes and of other parts of the equipment handling the material at pressures below atmospheric; also, the condensing water contains air in solution when at atmospheric pressure, and in direct-contact condensers much of this air is released under the diminished pressure and increased temperature in the condenser and is added to that which enters in the other ways just mentioned.

(c) The **effect of the air present** in the vapor being condensed has an important influence on the condenser pressure, for according to *Dalton's Law*, the total pressure of the mixture is the sum of the partial pressures of the air and vapor at the existing temperature. The influence of the air present is best appreciated from an example:

Assume the temperature of the exhaust mixture entering the con-

³ See instructions in A.S.M.E. Test Code for Condensers.

denser to be 96.8 deg. fahr. Then, if only saturated steam were present at this temperature, Fig. 893 shows that its absolute pressure would be 1.75 in. Hg, corresponding to which its specific volume is 380 cu. ft. per lb. If, in addition, each pound of vapor has mixed with it $\frac{1}{4}$ lb. of air, this air will have the same volume and temperature as the vapor, and its pressure will be

$$p = \frac{wRT}{144V} = \frac{53.3 \times (96.8 + 460)}{4 \times 144 \times 380} = 0.135 \text{ lb. per sq. in., or } 0.274 \text{ in. Hg.}$$

Thus the total pressure of the mixture as it enters the condenser is $1.75 + 0.274 = 2.02$ in. Hg, the increase due to the air present being

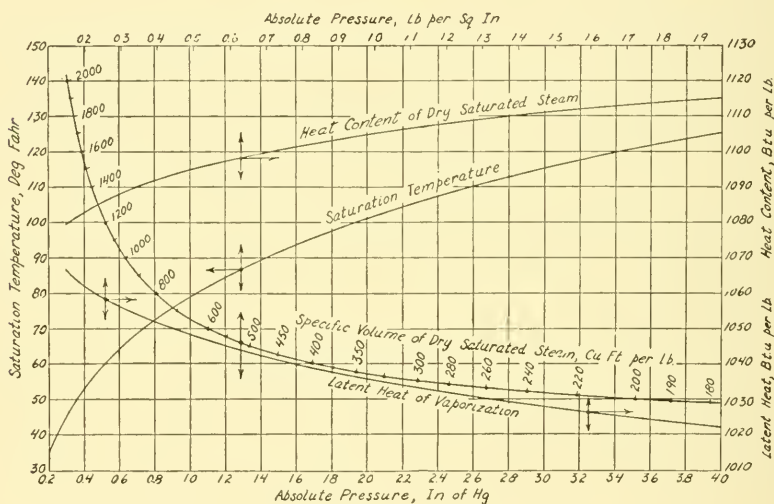


Fig. 893.—Properties of Saturated Steam at Condenser Pressures.

relatively quite large. (Had the temperature and pressure of the mixture been given initially, the partial pressures and relative weights of the air and vapor present could have been found by reversing the foregoing procedure.)

(d) In carefully maintained plants the air infiltration into the steam system is quite small; hence, as the heat content of the exhaust steam is very great compared with that of the entrained air, and as the relative weight of this air is negligible, it is usually sufficiently accurate, as far as the prime mover is concerned, to assume that only the vapor is the working substance. But as the mixture proceeds through the condenser the weight percentage and the partial pressure of the vapor

become lower and those of the air larger, the final values attained at the air outlet depending upon the degree of cooling or "refrigeration" encountered. Thus, if at the air outlet the absolute pressure of the mixture is 2 in. Hg and the temperature has been reduced to 86 deg. fahr. the partial pressure of the vapor is 1.25 and that of the air is 0.75 in. Hg. If, however, the mixture has been cooled to 70.4 deg., and the total pressure remains the same, the vapor pressure becomes 0.75 in. and the air pressure 1.25 in. (i.e., the partial pressures of the two substances are exactly reversed) and much less vapor will be removed with the air. With the 86-deg. temperature the air-to-steam ratio is 0.95 by weight, and with 70.4 deg., this ratio is 2.7.

(e) The air entering a condenser is detrimental not only because it increases the condenser pressure, but also because its removal, together with its entrained vapor, is costly, and it seriously hinders the transfer of the heat from the steam to the water. Thus, every precaution should be taken to reduce to a minimum the amount of air entering the condenser, but complete elimination is not possible. If this air is allowed to accumulate within the condenser it will gradually increase the back pressure on the prime mover, hence it must be removed as rapidly as it is brought in. Before the mixture can be discharged from the condenser, however, its pressure must be raised to a value somewhat above atmospheric, and this is done by means of an air compressor or equivalent device. The air is removed from surface condensers and some kinds of jet condensers by means of so-called *air ejectors*, *air pumps*, or *vacuum pumps*, which will be discussed in Sect. 543. With jet condensers of the kind from which the water is discharged at high velocity, the air may be entrained by the water and be ejected therewith, in which case a separate device for air removal is not needed.

525. The Heat Content of Exhaust Steam.—(a) The heat content of the steam entering the condenser from the prime mover in operation usually must be determined indirectly because the quality of this rapidly flowing steam cannot readily be directly determined. To estimate this heat content indirectly, an energy balance for the prime mover must be derived by equating the energy supplied to the prime mover in some unit of time to the energy leaving in all forms in the same time. If the heat content and weight of steam supplied the prime mover, the heat content and weight of any steam extracted, the energy delivered by the shaft, and the energy lost by heat transfer from the prime mover are all known, the energy rejected to the condenser in unit time may be estimated from the energy balance. Then, with this energy rejected to the condenser determined, the heat content of the

exhaust steam may be found if the weight of steam entering the condenser in unit time is known.

(b) **Typical values** of the heat content of the steam leaving large modern steam turbines are obtainable from the condition curves (broken lines) in Fig. 343 (page 114). Even when considering turbines that may have slightly different values of the stage efficiency than those for which these curves were drawn, this figure may still be used to advantage for quick estimates. From these curves it may be noted that for an

exhaust pressure of 0.5 lb. per sq. in., the heat content of the exhaust steam would often be near 950 B.t.u. per lb.

526. Amount of Condensing Water Needed.

—(a) The weight of water required to condense a given weight of steam under specified conditions may be found for all types of condensers, whether direct contact or surface, from the equation that expresses the energy balance for the condenser. Neglecting the effects of the

presence of air, the changes in velocity and elevation of the steam, and the heat lost to the atmosphere, this equation simply states that the energy given up by the exhaust steam equals that absorbed by the condensing water; or

$$w_s(h_s - h_c) = w_w(h_2 - h_1), \quad . \quad . \quad . \quad . \quad (638)$$

where

w_s and w_w = the respective weights, in lbs., of steam and condensing water flowing through the condenser in a unit of time, usually the hour.

h_1 and h_2 = the respective heat contents of the condensing water entering and leaving the condenser, in B.t.u. per lb.

h_s = the heat content of the steam entering the condenser, in B.t.u. per lb.

h_c = the heat content of the condensate leaving the condenser, in B.t.u. per lb.

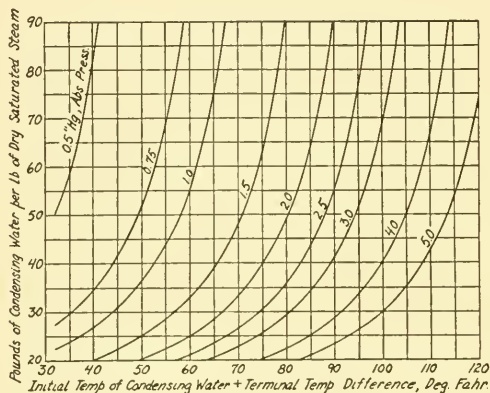


FIG. 894.—Relations between the Water-to-Steam Ratio, Exhaust Pressure, and Temperature of Condensing Water plus Terminal Temperature Difference (Dry Saturated Steam).

Then, the weight of condensing water per unit weight of steam becomes

$$\frac{w_w}{w_s} = \frac{h_s - h_c}{h_2 - h_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (639a)$$

or, very closely,

$$\frac{w_w}{w_s} = \frac{h_s - (t_c - 32)}{t_2 - t_1}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (639b)$$

$$= \frac{h_s - (t_c - 32)}{t_s - (t_1 + \theta_b)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (639c)$$

where t_1 and t_2 = the respective temperatures of the condensing water entering and leaving the condenser, in deg. fahr.

t_c = the temperature of the condensate leaving the condenser, in deg. fahr.

t_s = the temperature of the exhaust steam (assuming no superheat), in deg. fahr.

$\theta_b = t_s - t_2$ = the terminal temperature difference.

(b) With **surface condensers**, the temperature of the condensate (t_c) is nearly equal to the exhaust steam temperature (t_s). Exact equality is desired, especially if the condensate is to be used directly as feedwater. Assuming that $t_c = t_s$ and that the steam being condensed is dry saturated and air free, then $h_s - h_c = r_s$, the latent heat of the exhaust steam. For such conditions, the values of w_w/w_s for any given back pressure, injection water temperature, and terminal difference can be determined directly from Fig. 894. This figure is useful also as an aid in visualizing the rapidity with which w_w/w_s must be increased with reduction in back pressure and with elevation of cooling water temperature. If the steam is wet and the other conditions are as before, the values from Fig. 894 may be used after correcting them for the quality. If $t_c < t_s$, then Eqs. (639a) and (b) must be used instead of the curves. The best values of θ_b and of $(t_2 - t_1)$ to employ with a surface condenser are dependent on the operating conditions (including the temperature of the cooling water and the condenser pressure) and on economic considerations which will be discussed later. Although at times it is possible to reduce θ_b to 5 deg. fahr. with clean tubes and air-free steam, it is safer to allow a terminal temperature difference of at least 10 deg. The average value of $(t_2 - t_1)$ for a number of large two-pass surface condensers in operation during 1928-29 was about 9 deg. fahr., with ± 2 deg. variation, and it was a little less with single-pass condensers.⁴

⁴ Proc. N.E.L.A., 1929, p. 1458.

(c) With **jet condensers**, since $t_c = t_2 = t_{\text{mix.}} = t_s - \theta_b$, Eq. (639b) reduces to

$$\frac{w_w}{w_s} = (h_s - t_{\text{mix.}} + 32) \div (t_{\text{mix.}} - t_1) \quad (640a)$$

$$= (h_s - t_s + \theta_b + 32) \div [t_s - (t_1 + \theta_b)] \quad (640b)$$

As θ_b is small compared with h_s in the numerator of the last equation, Fig. 894 can be used also to indicate, with an error of but a few per cent, the values of the ratio (w_w/w_s) for condensers of this type even though

the conditions are not quite the same as those for which the figure was drawn. For more accurate determinations, the equations, rather than the curves, should be used.

Terminal temperature differences (θ_b) obtained in practice with large jet condensers operating at from three-quarters to full load, when plotted with respect to injection water temperatures, nearly all fall within the curved band shown⁵ in Fig. 895. It is seen that the higher the temperature of the injection water the smaller is the value of θ_b found in practice. The smaller values of this quantity are also obtained with steam that is comparatively free from entrained air.

(d) The actual **measurement of condensing water** supply-rates

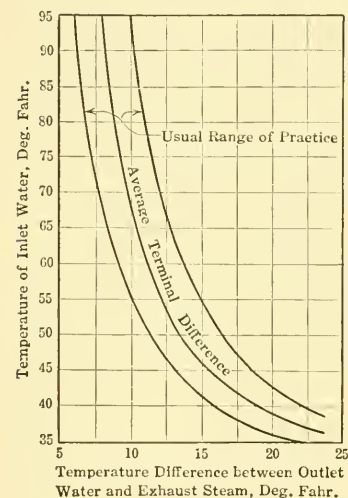


FIG. 895. — Relation between the Terminal Temperature Differences and Injection Water Temperatures with Large Jet Condensers.

is often difficult because of the great quantity of water involved. Usually this quantity can be estimated with sufficient accuracy from Eq. (639b). However, it is often simpler to estimate the amount of condensing water for a known rise in its temperature directly from the amount of energy that is absorbed by this water; and this amount of energy may be found for many cases by a simple method that will now be given.

(e) **Reasonable estimates** of the amount of energy absorbed by the

⁵ Report of Prime Movers Committee, N.E.L.A., 1923, Part B, p. 95. Based upon data obtained from jet condensers serving turbines of from 10,000- to 45,000-kw. capacity.

condensing water per kilowatt-hour output of large turbine-generators having an exhaust pressure of 1 in. Hg abs. may be quickly obtained with very little calculation by using the lower curves in Fig. 349 (page 130). These curves give, for a wide range of throttle pressures and temperatures, the energy consumption ⁶ in B.t.u. per kw-hr. of turbine-generator output, for four different cycles, provided the generator output equals 94 per cent of the energy delivered by the turbine blading. Consequently for such a unit, by neglecting the small amount of radiation from the turbine and condenser, there is obtained the following equation, having each term in B.t.u. per kw-hr.,

$$\left(\begin{array}{c} \text{Energy absorbed by} \\ \text{condensing water} \\ \text{for ex.p. of 1 in. Hg abs.} \end{array} \right) = \left(\begin{array}{c} \text{Energy consumption} \\ \text{of turbine-generator} \\ \text{from Fig. 349} \end{array} \right) - 3630. \quad (641)$$

The constant, 3630, is obtained by dividing 3413 by 0.94, and thus represents the energy delivered to the shaft of the turbine by the blades. Note that 4 feedwater heaters are used for the curves in (b), (c), and (d), of Fig. 349, and thus for a smaller number of heaters the corrections obtainable from the curves in Figs. 338 and 339 (page 109) should be applied, if the accuracy of the other data available in any given case justifies it.

527. Jet or Direct-Contact Condensers—General.—(a) Condensers of this class may be subdivided into the following types: (a) the *barometric*, or high-level jet, (b) the *low-level jet*, and (c) the *ejector type*. They differ mainly as to the methods employed in removing the condensate and cooling water from the apparatus.

(b) In general, compared with surface condensers, the types now under consideration have the following **advantages**: they (1) are less complicated, (2) are cheaper, (3) occupy less building space, (4) have lower maintenance expense, (5) do not require separate condensate pumps, (6) use less cooling water, and (7) this water may be of a character unsuitable for use with the other type. However, as the steam mixes directly with the cooling water in jet condensers, there are the following **disadvantages**: (1) the discharge is unsuitable for use as feedwater, unless the cooling water is of the requisite purity; (2), if air-removal apparatus is used, it must be of larger capacity and will require more power for its operation; and, (3) the cost of the water pumping against the relatively high heads encountered may be a serious item of expense.

⁶ This term always means that the turbine is not charged with the energy carried by the condensate leaving the hotwell, or the last regenerative feedwater heater. (See Part I, pages 320 and 352.)

528. Barometric Condensers.—(a) A condenser of this type consists essentially of a condensing chamber (head or cone), wherein the steam is brought into direct contact with the condensing ("injection") water, and a discharge pipe (tail pipe or barometric tube) of such length that, despite the vacuum in the condensing chamber, the water from the latter will be discharged through this pipe by *gravity*. The bottom of the tail pipe is submerged in the water of the hotwell or sump into which it discharges, and, as a 34-ft. head of "solid" water corresponds to standard atmospheric pressure, this pipe is commonly made of such length that the condenser head is from 35 to 40 ft. above the hotwell water level.

(b) In the simple arrangement of barometric condenser shown in Fig. 896, the water entering the head *B* issues downward in a thin annular sheet from around the central cone, condenses the steam passing through the cone, and the mixture discharges through the throat or neck (*N*) with sufficient velocity to carry with it the non-condensable gases and vapor. The flow of water is regulated by means of the hand wheel *H*, and an atmospheric relief valve, *A*, is provided to permit exhausting into the atmosphere, i.e., operating non-condensing, if the condenser is out of commission. The height of the water level (*L*) in the tail pipe depends upon the degree

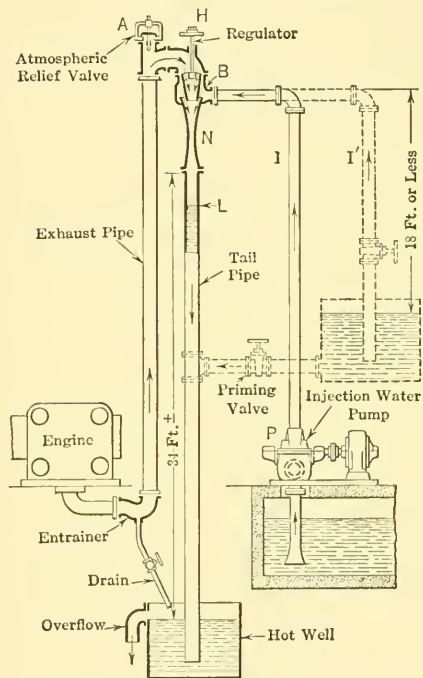


FIG. 896.—A Simple Barometric Condenser.

of vacuum maintained and on the density of the column of fluid. If the source of the cooling water is not over 18 ft. below the injection water inlet, then, under ordinary conditions, the water will flow into the condenser without assistance after the vacuum has once been established. Greater differences in level necessitate using a pump for supplying the cooling water to the condenser.

(c) Commercial condenser heads are made in a great variety of arrangements, which differ principally as to methods of bringing the steam and water into contact and of disposing of the air. In the form

of head just considered, the water, vapor, and air have parallel flow, and the air is removed solely by entrainment; hence, such arrangement is ordinarily suitable for obtaining only a moderate vacuum, usually not over 26 in. Hg. Better results are obtained by separating the air and removing it independently from the head, preferably after further cooling, using the counterflow principle. Fig. 897 shows an arrangement which meets these requirements, the air being removed from the top of the shell by means of an air pump after having passed upward through a spray of descending cold water.

(d) Because of their great height, barometric condensers are often placed out-of-doors, require the use of long exhaust piping (which adds to the cost, pressure loss between prime mover and condenser, and the possibility of air leaks), and commonly have high operating expense for pumping.

(e) In some interesting special applications, barometric condensers have been *employed as feedwater heaters*. In such cases the feedwater is used as the injection water and it is heated by condensing the exhaust from auxiliary apparatus, on which the back pressure is maintained at a pressure sufficiently high to give the feedwater the desired temperature.⁷

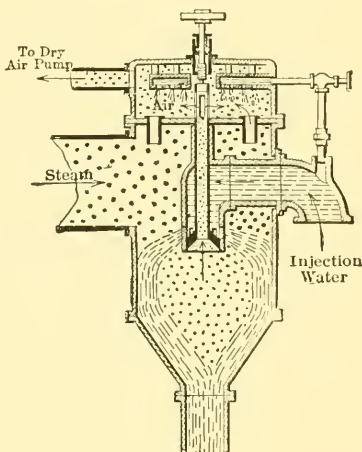


FIG. 897.—Head of a Counterflow Barometric Condenser.

529. Low-Level Jet Condensers.—(a) Direct-contact condensers of this type have condensing chambers resembling somewhat those of barometric condensers, but instead of the long tail pipes of the latter, they have water-removal pumps or tail pumps substituted for them. When these pumps also serve to remove the air, such condensers are commonly designated as **low-vacuum jet condensers** because of the inherent air-removal limitations of such pumps. When the air is removed by separate air ejectors or air pumps it is possible to obtain lower back pressures, and when so equipped the condensers are usually referred to as **high-vacuum jet condensers**. Because of their relatively small height, low-level jet condensers are commonly placed directly under the prime movers they serve and hence do not require additional floor space. They are extensively used in localities where the supply of cooling water is of a character suitable for use as feedwater or where

⁷ Trans. A.S.M.E., Vol. 43 (1921), p. 487 (Colfax), and p. 500 (Detroit Edison).

reasonably good feedwater can be obtained from other sources at little expense: also they are the kind commonly selected when the condensing water available is contaminated by acids or other foreign matter which renders it unsuitable for use in the tubes of condensers of the surface type.

(b) A typical arrangement of high-vacuum condenser of the type under consideration is shown in Fig. 898. The steam and entrained air enter this condenser at the top, and in the *condensing cone* passes through a spray of condensing water which, with the condensate, collects in the *hotwell* from which the mixture is discharged by a *removal pump* of the

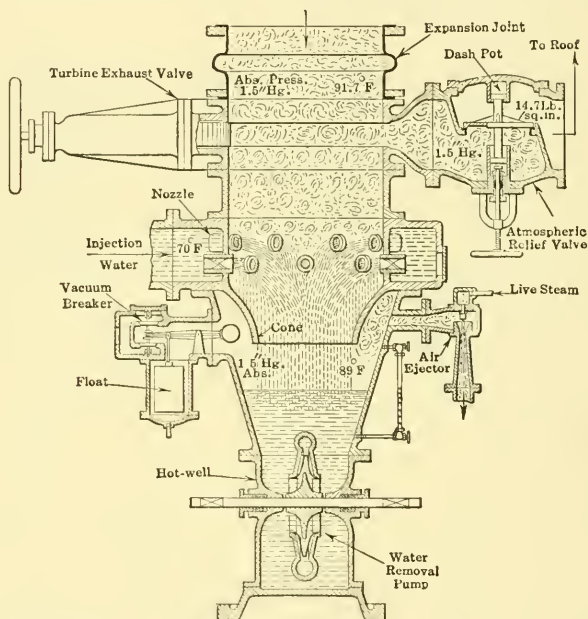


FIG. 898.—Section of Typical Low-Level High-Vacuum Jet Condenser.

centrifugal type. The air-vapor mixture remaining after the condensation process is completed passes under the cone through the coldest region in the condenser body, and is evacuated by means of a steam-jet *air ejector*. The condensing water enters the condensing cone through *spray nozzles* and is supplied by an *injection-water pump*, unless the head is not over (say) 18 ft. In the latter case this water will flow into the condenser without assistance after the vacuum has once been established. The float-operated *vacuum breaker* is a safety device which functions so as to open to the atmosphere when the water level becomes dangerously high, and might, but for this device, flood the prime mover. The *atmospheric relief valve* opens automatically and safeguards the con-

denser shell from excessive pressure when the vacuum is destroyed; it functions when the supply of injection water fails, when the vacuum breaker operates, or when the *exhaust valve* is closed so that the condenser can be repaired while the turbine operates non-condensing. The exhaust valve is often omitted, and the *expansion joint* is not always required.

(c) In the **typical installation** of a condenser of this type, shown in Fig. 899, the following should be noted: The condenser is coupled directly to the turbine exhaust opening, except for the interposition of the expansion joint and the exhaust valve. The injection-water pipe contains a regulating valve, and its inlet end is flared and submerged 3 ft. or more so as to preclude the possibility of air descending to it from the surface of the intake water. The removal pump has a dual drive so that it can be operated either by an electric motor or a steam turbine. (Dual drives are required only in special cases.) This pump receives the mixture of condensate and cooling water at about condenser pressure and discharges it against atmospheric pressure and the other resistances encountered; hence such condensers consume a considerable amount

of energy as compared with a barometric condenser, which has gravity discharge, and with surface condensers, the condensing water pumps of which function against a relatively low head, and only a comparatively small amount of condensate is discharged against the higher head. The discharge pipe from the removal pump contains a check valve which prevents the water in the discharge tunnel from flooding the condenser and turbine should this pump cease to function while the condenser is under reduced pressure.

530. Ejector Condensers.—From condensers of this type the injection water, condensate, and air are ejected without the use of either discharge pumps or barometric columns, the mixture being removed by having the injection water flow through the condenser with sufficient kinetic energy to discharge the mass against atmospheric pressure.

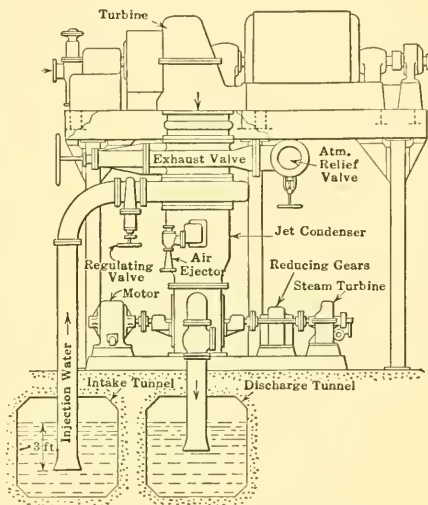


FIG. 899.—Typical Installation of Low-Level High-Vacuum Jet Condenser.

High-vacuum ejector condensers are usually of the multi-jet low-level arrangement, a typical installation of which is shown in Fig. 900. The injection water, which is delivered to the nozzles of this condenser at a gage pressure of about 9 lb. per sq. in., issues in converging jets which condense the steam and entrain the air while passing at high velocity through the combining tube. The mixture, under the combined effect of the external water pressure, the vacuum within the condenser shell, and gravity, is discharged through the short tail diffuser pipe into the hotwell. In flowing through the gradually enlarging tail pipe the jet velocity progressively decreases, and the pressure correspondingly

increases until it is sufficient to overcome the atmospheric pressure. The air brought in with the injection water remains permanently in solution and that entering with the steam is quite effectively removed by the injector action; thus a high vacuum can be maintained with this simple apparatus without the assistance of separate air pump. Should the water supply fail, the existing vacuum would draw water from the hotwell into the condenser chamber, whereupon the float-actuated vacuum breaker would operate to admit air and thus destroy the vacuum. An additional safeguard against flooding the condenser and the prime

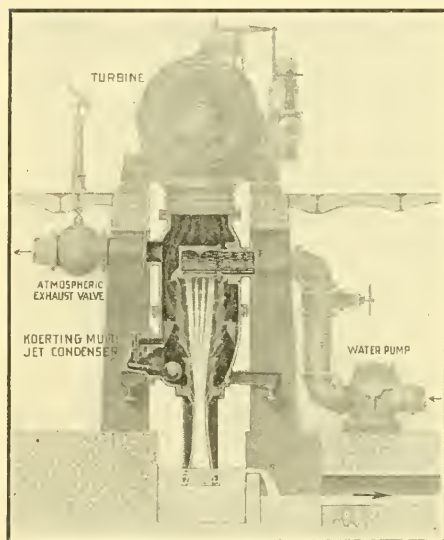


FIG. 900.—Multi-Jet Condenser of the Ejector Type (Schutte-Koerting).

mover is afforded by limiting the amount of water which can flow back from the hotwell before the end of the discharge pipe becomes uncovered. These condensers possess the advantages of the ordinary or conventional low-level jet condensers and in addition do away with the air pump and the removal pump. This simplification, together with the low initial and maintenance cost, ease in starting and stopping, and adaptability to special conditions, may offset the greater amount of injection water that may be required by them.⁸ Low-vacuum con-

⁸ For test data on a low-level multi-jet condenser see Report of Prime Movers Com., N.E.L.A., 13-22, 1922, p. 23, or Gebhardt's *Steam Power Plant Engineering*, p. 513.

condensers of this type commonly have a single jet of water flowing through the combining tube. Ejector condensers can also be used with barometric tail pipes, as in Fig. 896.

531. Surface Condensers—General.—(a) Compared with usual direct-contact condensers, surface condensers have the following **principal advantages**: (1) the condensate, which is quite completely de-aerated and free from scale- and mud-forming impurities, can be used as feedwater, (2) the recoverable heat of the condensate can be quite fully regained with such use, (3) the same feedwater is used repeatedly, thus avoiding the expense for new water and its treatment (which is important only when suitable make-up water is difficult to obtain or is costly), (4) higher vacuums can be maintained under fluctuating loads and varying temperature of circulating water, (5) better vacuums are generally obtainable with air pumps of smaller capacity, (6) less power is required for operating condenser auxiliaries, (7) there is no possibility of the condensing water flooding and wrecking the prime mover, and (8) there is less liability of losing the vacuum, as a decrease in the latter does not affect the flow of cooling water to the condenser. In addition, the condenser may be credited in a large measure with the benefits accruing in the steam-generating plant from using the condensate as feedwater; such use may cause the steam generators to have higher efficiency and capacity because of having cleaner tubes, may decrease the frequency of shutdowns for cleaning, and may decrease internal deterioration.

(b) The **principal disadvantages** attributable to surface condensers are: (1) their greater initial cost, (2) the greater space needed for them and for tube removal and replacement (also higher head room is usually required to provide adequate supply-head for the hotwell pumps), (3) the greater attention and maintenance cost involved, and (4) their unsuitability for use with circulating water containing foreign matter which will cause rapid deterioration, scaling, or clogging of the tubes. The upkeep cost is mainly for cleaning, for keeping tight the multitudinous joints at the tube ends, and for replacement of deteriorated condenser tubes.

(c) The **classification** of surface condensers can be made in many ways, only a few of which will be given here. Such condensers are usually designated primarily as *high-vacuum* condensers, which produce relatively low absolute pressures within them (e.g., 2 in. Hg or less), or *low-vacuum* ones, which are usually limited to producing back pressures not below 3 or 4 in. Hg; as *single-pass* or *multi-pass* condensers, depending on whether the circulating water flowing through the tubes crosses the steam space one or more times; as *horizontal* or *vertical*

condensers, in accordance with the position of the tubes; and as *small* or *large* condensers. Most surface condensers are of the high-vacuum type with horizontal tubes and are of either the single-pass or two-pass arrangements. Discussion will be confined to such condensers unless otherwise indicated. The *size* of a surface condenser is commonly expressed in terms of the number of square feet of effective tube surface exposed to the steam. Small condensers are often of the *unit type* in which the condenser and its auxiliaries are of standardized design constituting an integral unit. Small condensers, and low-vacuum ones, ordinarily do not require some of the niceties of design necessary with large high-vacuum condensers. Large condensers are usually of *individual design* and are combined with separately selected auxiliaries, each chosen with respect to the specific requirements of the particular installation. Low-vacuum condenser units can be of simpler and less costly design than high-vacuum ones. Other classifications will appear in connection with the more detailed discussion given in subsequent sections.

(d) The **selection** of the surface condenser design best suited to specific requirements involves an economic analysis based upon such factors as (1) the space limitations or requirements, (2) the adequacy of the water supply, (3) the pumping heads and resistances, (4) the character of the loading, (5) the hours of operation per year, and (6) the cost of auxiliary energy. Usually the single-pass arrangement is justified when there is a liberal supply of condensing water, combined with low pumping head and low energy cost, whereas the two-pass arrangement would be the more suitable one with conditions reversed. In general, the heavier the loading and the longer the period of operation per year, the greater is the expense justified in producing a better vacuum. Further consideration of the selection and design of surface condensers will be found in Sect. 538, page 910.

(e) The **main desiderata** for the **steam side** of a surface condenser are:

(1) The *distribution of the steam flow* within the condenser should be such as to make all parts of the condensing surface equally effective; also the resistance to this flow should be the least possible.

(2) The *effectiveness of the condenser surface* should be maintained at a maximum by having the surface always free from blanketing masses or films of air, water, oil, or other material having low heat-conductivity. The films are minimized by having the steam flow over all parts of the tube surfaces at high velocity, and the blanketing is avoided by properly arranging the steam and air passages through the nests of tubes and by making adequate provision for air and water removal.

(3) The *final air-vapor mixture* should be separated from the main

mass within the condenser and should be cooled independently before delivery to the air pump.

(4) The *condensate* should be deaerated, and should be maintained at the temperature of the exhaust steam by keeping it from contact with the colder surfaces, gases and vapors; or if cooled in such manner, it should be brought again into intimate contact with the exhaust steam.

On the **water side** of the condenser:

(1) The *distribution of the condensing water* should be such that all tubes are equally effective.

(2) The *flow of the water* through the condenser should be with the least friction and static head, and with the application of the counter-current principle as far as is expedient, the coldest water always being supplied to the air-cooling tubes.

(3) The *circulating water* should be of a character that will not cause deterioration of the condenser tubes, nor form deposits on the tube surfaces or within the water boxes of the condenser.

Important **structural requirements** of a surface condenser (in addition to having suitable proportions, materials, strength, and low cost of parts) are:

(1) The *joints at the tube ends* should be leak proof so as to preclude contamination of the condensate by leakage of condensing water through them.

(2) *Provision for inequalities of expansion* should be made for such differences as may occur between the tubes and the condenser shell, and between the latter and the attached auxiliaries and piping.

(3) The *character of the condenser support* and the *method of the attachment* to the exhaust flange of the turbine should be such as not to induce undue stresses or distortions in the latter due to variations in loading or to differences in expansion.

532. Typical Surface Condensers.—(a) The arrangement and elements of a typical double-pass **unit-type condenser** are shown in Figs. 901 and 902, in which many of the parts are labeled, the figures being largely self-explanatory. The condenser is located immediately below the outlet of the prime mover and, since it is small, the condensate and circulating pumps can be placed directly under it without requiring an excessive amount of head room. The condenser and pumps are of standardized unit-design, each manufacturer having his own designs for different sizes and conditions. As in all surface condensers, the circulating water is injected at the lowest point in the intake water-head of the condenser and is discharged at the highest point so that all tubes will be effective whenever water is overflowing. The steam enters the shell at the top and flows downward between the tubes, the condensate

gravitating to the hotwell and condensate pump below. The final air-vapor mixture is separated from the main mass near the bottom of the

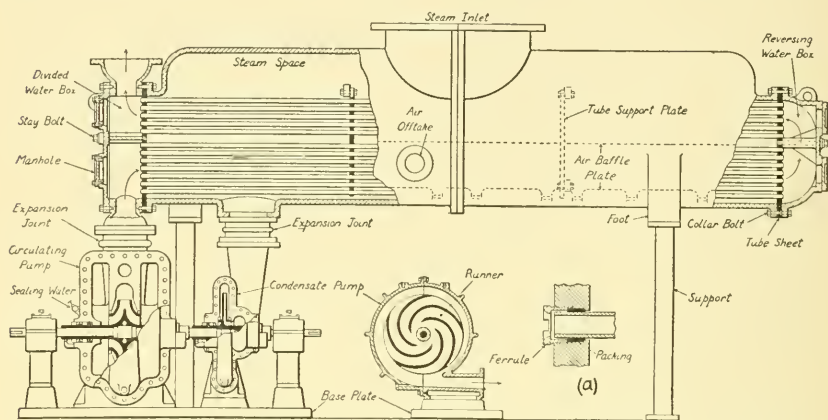


Fig. 901.—Typical Unit-Type Surface Condenser.

(See Fig. 902 for Transverse Sections through Circulating Pump, Half of Water Box and Half of Shell.)

shell, passes under a baffle plate, and is cooled by a special group of air-cooling tubes before passing through the side outlet to the air pump or ejector (not shown in the figure). The condensate and circulating pumps are both of the centrifugal type and their impellers are mounted on the same shaft, hence are operated from a common drive. The pumps are connected to the condenser by means of flexible expansion joints. In a single-pass arrangement the partition in the intake water box would be omitted and the circulating water would be discharged from the top of the other water box. One method of joining the tubes to the tube sheets is shown at (a) in Fig. 901; for various methods employed see Sect. 534(b). A typical installation of one of these condensers is illustrated in Fig. 903.

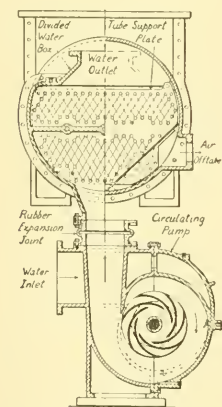


Fig. 902.—Transverse Sections of Condenser Shown in Fig. 901.

(b) A **simple low-vacuum unit**, such as is suitable for maintaining pressures not below 4 in. Hg, is shown in Fig. 904. It is of the double-pass type and has steam-driven reciprocating pumps which constitute an integral part of the unit. The pump plungers and steam piston are all on the same rod. The condensate pump serves also as the air-removal

pump, hence is termed a **wet vacuum pump**. Such units are often used in small installations requiring low cost and not demanding low condenser pressures. With them the temperature of the air entering the pump is higher and the condensate is cooled more than when the air

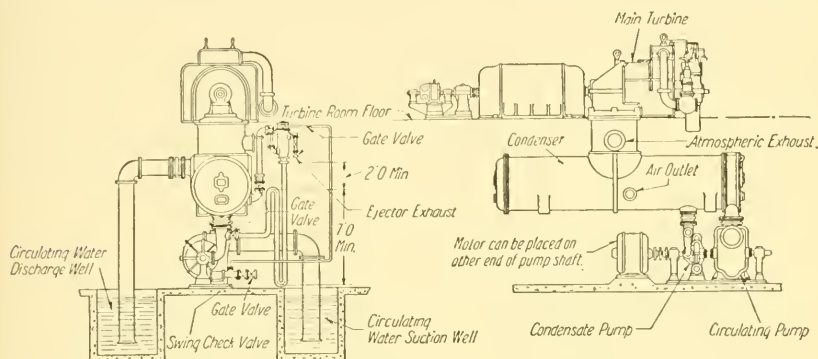


FIG. 903.—Typical Installation of Unit-Type Surface Condenser. (Westinghouse.)

is handled independently by a separate **dry vacuum pump** or air ejector.

(c) One illustration of a **large condenser of individual design** and double flow arrangement is shown in Fig. 905. In this example the

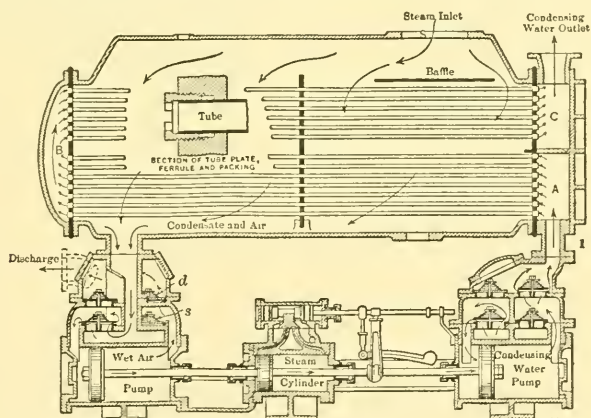


FIG. 904.—Small Low-Vacuum Surface Condenser Unit.

water boxes are divided vertically so that either half of the condenser can be shut down for cleaning, repairs, or adjustments on its water side while the other half continues in operation although with a slightly reduced vacuum.

A **typical installation** of a divided surface condenser of individual design, with pumps in duplicate and separately driven, is shown in Fig. 906. The condensing water for this unit comes from the *sluice*, or fore bay, shown at the left of the figure, passes through a *trash rack*, *sluice valve*, and *traveling screen* (all of which can be blocked off by logs for repairs), and flows through a concrete-walled *intake tunnel* to the circulating pumps, the suction pipes of which are immersed to a depth sufficient to preclude the possibility of air being sucked in with the intake water, such air descending from the water surface. The circulating water from each half of the condenser is discharged into an *outlet tunnel*.

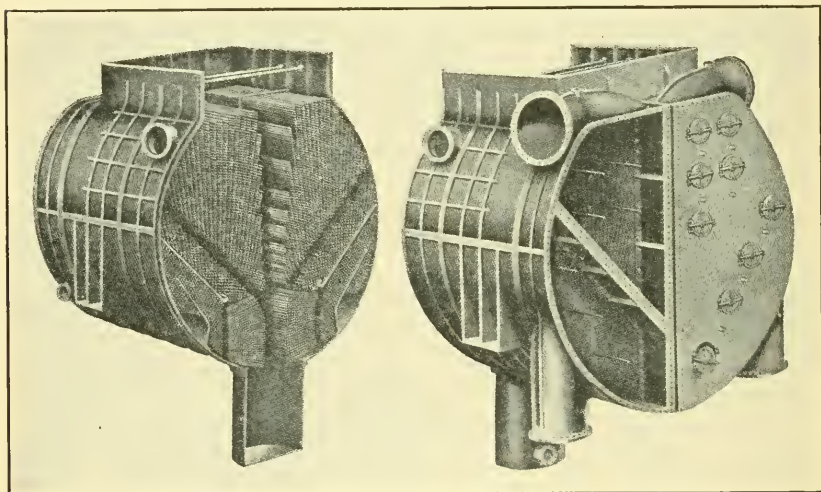


FIG. 905.—Large Condenser of Individual Design. (C. H. Wheeler Co.)

The condensate pump delivers its water to a small surface condenser (S) in the steam-jet air-ejector system for the purpose of condensing the ejector steam and recovering the heat of condensation thereof. This water then goes either direct to the feedwater system or to a storage tank. The condensate from the ejector condenser is drained to the hotwell through a sealing loop. All valves in the circulating system can be closed either by hand or by remotely controlled motors, in this case electric; often hydraulically or pneumatically operated valves are similarly used. Flexible expansion joints are placed between the condenser and the turbine, and between it and the larger piping. Often cast-iron pipes are substituted for the tunnels in small installations; and by having a suitably arranged slide valve in the top of the water box, a single discharge pipe may be used instead of two.

(d) In localities where the condensing water available contains foreign material (leaves, sticks, etc.) and is difficult to screen, the condensers are commonly so arranged that the flow of water through them can be reversed at frequent intervals so as to prevent the clogging of the tubes and water boxes. In regions where the level of the intake water varies greatly (as on the Ohio River, which rises 50 to 70 ft. in flood seasons), the turbine is located above high-water level, and the condenser is placed in a pit at about the normal level of the water so as to minimize the annual pumping cost. In such cases the condensers have their axes vertical, or nearly so, in order to reduce as much as possible

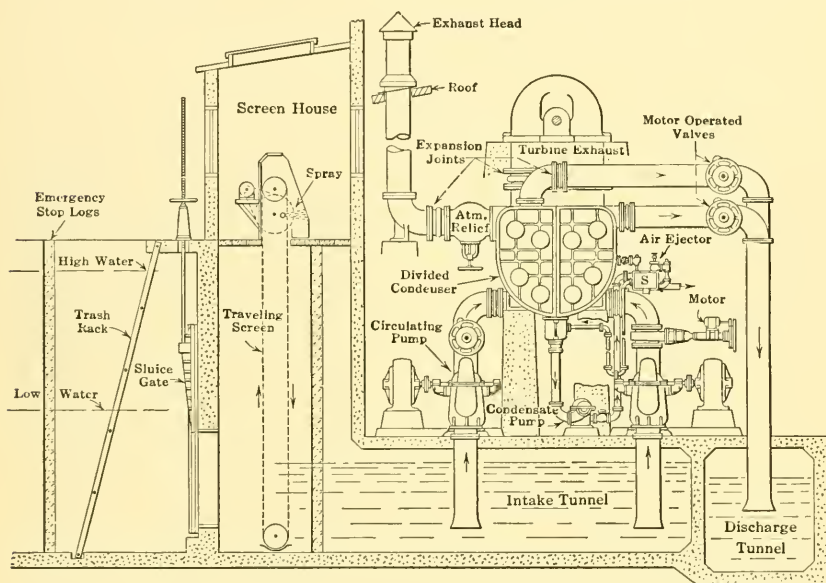


FIG. 906.—Typical Installation of Surface Condenser of Individual Design.

the width of pit needed for tube removal, and the exhaust pipe is very long and hence requires special provision for its expansion.⁹ Condensers with vertical tubes are also used when it is inexpedient to place them underneath the turbine. An example of an installation of vertical condensers is given in Fig. 316, page 80.

533. Tube and Hotwell Arrangements.—(a) With the **simple tube arrangement** shown in Fig. 904, there is a considerable drop in pressure from top to bottom of the condensing chamber (e.g., 0.1 to 0.4 or more in. Hg), the condensate formed on the upper tubes is cooled below the exhaust temperature when it passes over the lower tubes, the air is

⁹ See "Columbia Power Station," in *Power Plant Engineering*, Feb. 1, 1926, p. 193.

removed under unfavorable conditions, the steam is not distributed equally to all parts of the tube surface, the lower tubes are less effective

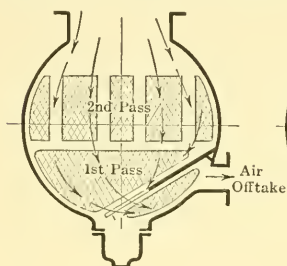


FIG. 907.

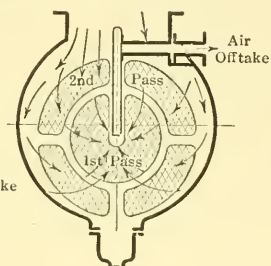


FIG. 908.

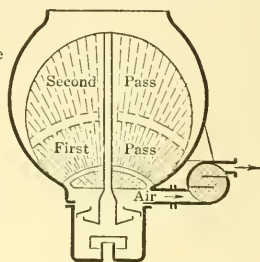


FIG. 909.

FIG. 907.—Tube Bank with Steam Lanes.

FIG. 908.—Radial Flow Arrangement.

FIG. 909.—Tubes Arranged in Converging Rows. (Foster-Wheeler.)

than the upper ones, and the ends of the tubes are less effective than the middle portions. The deeper the tube bank is the worse become these undesirable features, hence this simple arrangement is not commonly used for large condensers, nor is it suitable for small ones requiring the best results obtainable. With the arrangement shown in Fig. 902 the air removal is more effective, and this layout is satisfactory for high-vacuum condensers of small size in which the tube nests are comparatively shallow.

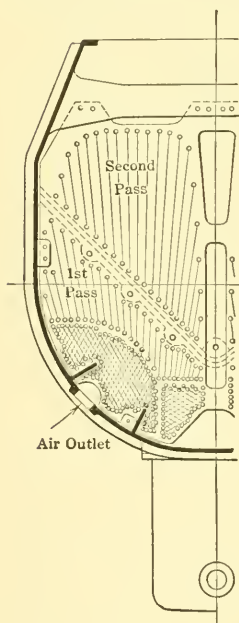


FIG. 910.—Dual Arrangement with Converging Tube Rows. (Elliott.)

(b) Typical transverse sections of large condensers, embodying in various degrees features intended to eliminate some or all of the undesirable characteristics of the simple arrangements just considered, are shown in Figs. 907 to 913, the first four being for two-pass condensers in which the coldest water passes through the air-cooling tubes, and the others for single-pass condensers. All show special provision for distributing the steam into the interior portions of the tube banks, so as to increase the effectiveness of all parts of the tube surface and decrease the total amount of surface required, and all provide for independent air removal.

In the tube arrangement shown in Fig. 907 the steam penetration is aided by means of *steam lanes* formed by omitting some

of the tubes. In the *radial flow arrangement* illustrated in Fig. 908 the exhaust steam forms a belt completely enveloping the tube bundle and, feeding the full periphery thereof, flows concentrically in comparatively short converging paths to the point of air removal which is located at the center of the first or coldest water-pass. In Figs. 905, 909, and 910 the steam distribution is aided by arranging the tubes in *converging rows*; and in Fig. 911 it is effected by using *variable tube spacing* in a heart-shaped shell, the tubes in the upper bank being farther apart than those in the lower banks. In Figs. 909 and 911 the air coolers are located outside of the condenser shell. The last figure shows an Ingersoll-Rand *single-pass* arrangement with a partitioned water box having two main injection nozzles, 1 and 2. Both nozzles are used in summer and only the upper one in winter, thus obtaining high water velocity through the effective tubes even when a reduced quantity of water is used in the colder season. Part of the circulating water passes first through the cooler and then through the bottom pass of the condenser. In the tube layout shown in Fig. 912, which is for one of the largest condensers so far constructed,¹⁰ not only are lanes provided for the steam but also *air-collecting lanes* are added for removing the final air-vapor mixture.¹¹

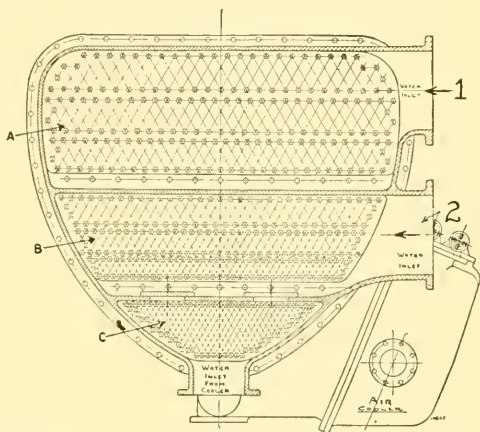


FIG. 911.—Layout with Variable Tube Spacing in Different Banks. (Ingersoll-Rand.)

¹⁰ This single-pass condenser serves a 160,000-kw. turbine in the Hudson Ave. Station of the Brooklyn Edison Co. It has 101,000 sq. ft. of tube surface (0.63 sq. ft. per kw.), and the water boxes are divided vertically, thus separating the condenser into two independent parts each served by its own circulating pump, the capacity of each pump being 73,250 gal. per min. The condenser contains 14,710 tubes, each $\frac{3}{4}$ -in. O.D. and with an effective length of 30 ft. The steam inlet is 20 ft. 10 in. by 20 ft. For further description of this and other large condensers see "World's Largest Condensers," in *Power*, May 31, 1932, p. 823; and for test results see "Performance of Two 101,000-Sq. Ft. Surface Condensers," by J. N. Landis and S. A. Tucker, *Power*, June, 1933, p. 308, and also in a paper given at the June, 1933, meeting of the A.S.M.E. at Chicago, Ill., by the same authors.

¹¹ For test of steam distribution in a somewhat similar tube arrangement, see "Condenser Design Based on Accurate Tests," by C. L. Waddell, in *Power Plant Engineering*, Feb. 15, 1931, p. 251.

Similar provision is likewise made in the arrangement shown in Fig. 913, and, in addition, *transverse division plates* are placed between the tube bundles to guide the steam and air and also to collect the condensate and discharge it into the vertical air lanes.¹² The real worth of the division plates is problematical, and in later condensers installed by the same company they have been omitted.

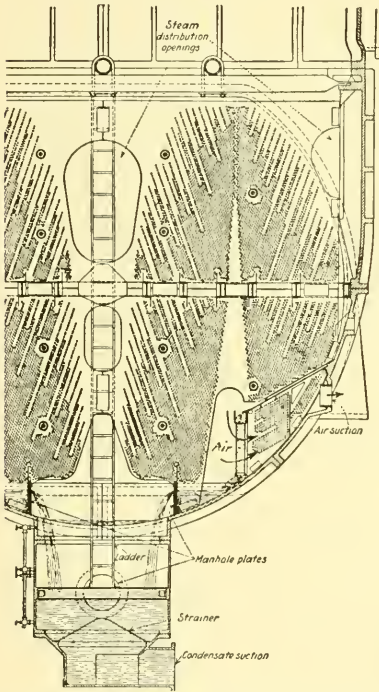


FIG. 912.—Large Single-Pass Condenser with Separate Lanes for Steam and Air, and with Condensate-Heating Hotwell.

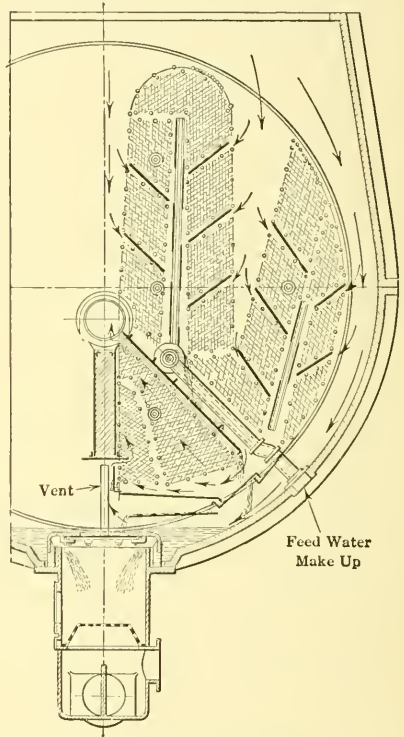


FIG. 913.—Condenser with Steam and Air Lanes, Division Plates for Collecting the Drip, and Condensate-Heating Hotwell.

(c) **Longitudinal distribution** of the steam over the full length of the condenser tubes is just as important as the transverse distribution. Provision is made for it by such schemes as having a large space or dome over the condenser tubes, by using a steam inlet nearly as long as the tubes, and by having large equalizing openings in the tube-supporting plates. One manufacturer (Ingersoll-Rand), on the other

¹² Described on p. 660 of the continued article on "The Trenton Channel Power Station," in *Engineering* (London), 1928, vol. 125.

hand, uses these plates as partitions to divide the condenser into several elements, each of which is, in effect, an independent condenser having its own steam inlet and air-pump and condensate outlets. **Equal water distribution** to all tubes is also of great importance.¹³

(d) **Condensate heating** to substantially exhaust temperature is accomplished in the layouts shown in Figs. 912 and 913, and in some earlier figures, by causing the exhaust steam which has not passed over tubes to come in contact with the condensate flowing to the hotwell or with that already collected therein. In such arrangements the water is delivered to the hotwell in cataracts and the exhaust steam is supplied thereto through large central or annular passages. Hotwells such as these function substantially as direct-contact feedwater heaters. **Deaeration** of the condensate is also accomplished quite effectively in combination with this heating process, and it is interesting to note that the condenser shown in Fig. 913 is used to deaerate also the feedwater make-up.¹⁴

534. Surface Condenser Parts.—(a) The shells of surface condensers have circular, flattened-circular, heart-shaped or other forms of cross-section, the kind selected depending on the particular design requirements or space limitations. Formerly the shells were almost always made of cast iron, but now they are often of welded construction built up of rolled steel plates. To prevent collapse due to the difference between the atmospheric pressure outside and the exhaust pressure within, the walls are usually ribbed and braced, and especially is this reinforcement necessary on the non-cylindrical portions. Small shells are made in one piece, but large ones are built up of separate sections, the size of which is limited by transportation facilities. *Water boxes* and covers, hotwells and manhole covers are usually of cast iron.

(b) **Condenser tubes** are generally $\frac{3}{4}$ in., $\frac{7}{8}$ in., or 1 in. outside diameter and 0.049 in. (No. 18 B.W.G.) thick, with the smaller sizes predominating, especially in the single-pass designs. Thicker tubes are used when rapid deterioration is expected. Tubes $\frac{5}{8}$ in. O.D. are sometimes used in small condensers, but are often difficult to clean. For most inland waters, tubes of *brass* (70 copper + 30 zinc) or *Muntz metal* (60 Cu + 40 Zn) are usually satisfactory. For sea water or water

¹³ See "Condenser Study Shows Bad Water Distribution," by J. J. Grob and N. Artsayooloff, in *Power*, Nov. 9, 1926, p. 702.

¹⁴ In the foregoing discussion no attempt has been made to cover all the constructions used by the various manufacturers of commercial condensers, the illustrations having been limited to only a few designs which emphasize important features. Most condenser manufacturers have a variety of designs, each suited to specific conditions. A study of manufacturers' catalogues and bulletins will prove of interest and value in supplementing the discussion here given.

contaminated by sewage or other detrimental foreign matter, the somewhat more expensive *Admiralty metal* (70 Cu + 29 Zn + 1 Sn) is commonly used. Other materials are sometimes employed for tubes, including copper alone, aluminum brass, arsenical copper, and aluminum alloys containing from 15 to 30 per cent nickel. Condenser tubes¹⁵ should conform to the A.S.T.M. standards.

(c) **Tube joints** were formerly almost always of the packed type formed by fitting the tube ends into recessed holes in the tube sheets and compressing around them rings of corset lace, fiber, or metallic packing, such material being placed in the recesses and being forced tight by means of threaded brass ferrules. Such joints allow for differences in expansion between tubes and shell. A joint of this kind is illustrated at 1 in Fig. 914; it will be noticed that the ferrule shown is

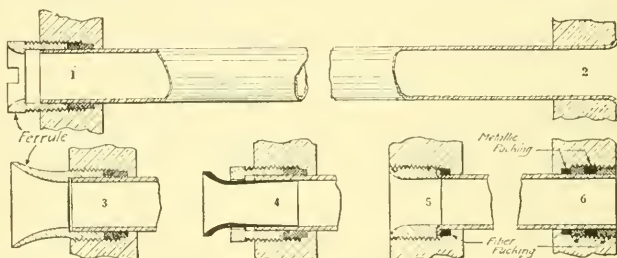


FIG. 914.—Some Examples of Tube Attachments.

- | | |
|------------------------|--------------------------|
| 1. Conventional Gland. | 4. "Flowrite" Bushing. |
| 2. Rolled Joint. | 5. "Smooth Flo" End. |
| 3. Belled Ferrule. | 6. "John Crane" Packing. |

slotted for screwing it in place, has an internal flange which prevents endwise displacement of the tube, and has a converging entrance. As leakage of condensing water through the tube joints contaminates the condensate, the tubes in modern condensers commonly have expanded or rolled joints at one end and packed joints at the other to allow for expansion, or they may have rolled joints at both ends, expansion differences in the latter case being provided for in other ways which will be discussed later. The entrance to and exit from the tubes should preferably be with stream-line flow so as to prevent eddies and consequent tube pitting, the cause of which will be explained when condenser maintenance is discussed. Some other examples of condenser tube attachments are also given in Fig. 914.

(d) **Tube sheets** are subject to collapse on account of the difference between the pressure of the condensing water on the one side and the

¹⁵ For data on tubes for condensers and feedwater heaters see Table LXVII.

exhaust pressure on the other. As they are honeycombed with closely spaced tube-holes, they have little inherent structural strength; hence special staying is necessary, especially when the sheets are large. They may be stayed by means of non-corrosive or sleeve-protected stay bolts extending between them and the heads of the water boxes, by fastening them to the partition plates in the water boxes, by struts extending from one tube sheet to the other through the steam space, or by tubes having rolled fits at both ends. When the tube ends are fitted with ferrules, the tube sheets are commonly made of rolled Muntz metal, but when they have expanded joints they are either of this material or of corrosion-resisting alloy steel (e.g., copper-bearing steel), the latter giving the stronger ligaments between the tube holes to withstand the rolling. Intermediate *tube-support plates* are of cast iron or alloy steel.

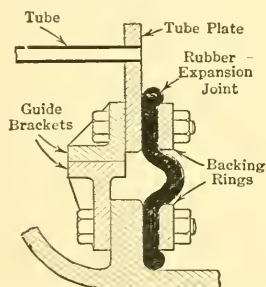


FIG. 915.—Floating Tube Sheet.
(Foster Wheeler.)

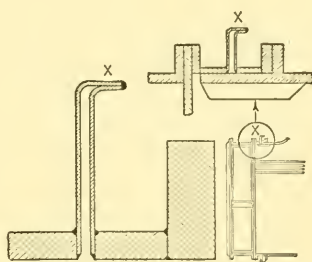


FIG. 916.—Expansion Joint in Shell.
(Allis-Chalmers.)

(e) **Provision for differences of expansion** of tubes with respect to the shell when rolled joints are used may be made in the following ways: If the tubes are sufficiently long, small in diameter, and thin, or if curved, they themselves provide the necessary flexibility, and in such case the rims of the tube sheets can be attached rigidly to the shell flanges. In other cases one tube sheet may be of the "floating type," which is flexibly connected around its edge to the shell by means of a diaphragm of corrugated sheet metal or of rubberized fabric, as in Fig. 915; or, if the tube sheets are rigidly fastened to the water boxes, one or both of the latter may be flexibly connected to the shell, or an expansion joint may be placed in the shell itself, as in Fig. 916.

535. Condenser Connections and Supports.—(a) The shell of a turbine is thin and quite flexible, and if distorted in any manner—as by loading due to the weight of the condenser and its contents, or by a relative displacement of a rigidly attached condenser, such as is caused by differences in expansion or by settling of foundations—the blades

may rub and be destroyed, serious stresses may be produced, or other difficulties arise. Also relative displacements between a condenser and its auxiliaries or piping rigidly attached to it are usually highly detrimental. Hence the methods of supporting the condenser and its related parts and of making connections between them in an installation require very careful consideration. In some arrangements it is

necessary to interpose a flexible expansion joint in the connection between the condenser inlet and the turbine exhaust flange; such joints are nearly always used between the condenser and its auxiliaries and between it and the injection and discharge pipes for the circulating water. Several

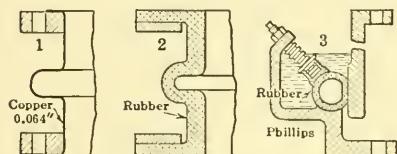


FIG. 917.—Some Examples of Expansion Joints.

types of **expansion joints** are illustrated in Figs. 852 and 917, and typical applications have been shown in Figs. 898, 899, and 906. In other arrangements the condenser's inlet flange is bolted directly to the turbine's exhaust flange and the condenser is freely suspended from the latter, part of its weight and that of its contents, however,

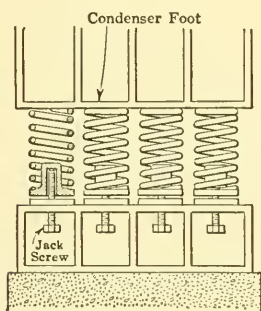


FIG. 918.—Spring Support for Condenser.

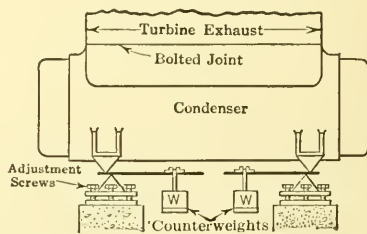


FIG. 919.—Counterweighted Condenser.

usually being borne by **spring supports**, as in Fig. 918, or by **counterweights**,¹⁶ as in Fig. 919.

(b) **Usual arrangements of condenser supports and connections to turbines** are as follows: (1) the condenser is bolted to its own foundation and connected to the turbine outlet by means of a single unbalanced expansion joint; (2) the condenser is bolted to the turbine's outlet

¹⁶ For discussion of spring and counterweight supports, see Proc. N.E.L.A., 1926, p. 983; counterweights are illustrated in Electrical World, May 1, 1926, p. 919.

flange and is suspended therefrom, part of its weight being carried by springs or counterweights, as already noted; (3) balanced, double opposed expansion joints are used; or (4) the condenser shell alone is suspended from the turbine exhaust flange, the water boxes being flexibly connected to the shell and independently supported. Each of these arrangements will be considered in more detail.

(1) **Using a single expansion joint** between turbine and condenser and bolting both of these elements to their respective foundations relieves the turbine of any loading due to the weight of the condenser and its contents. It is important to note, however, that with such arrangement the exhaust shell of the turbine is subjected to an unbal-

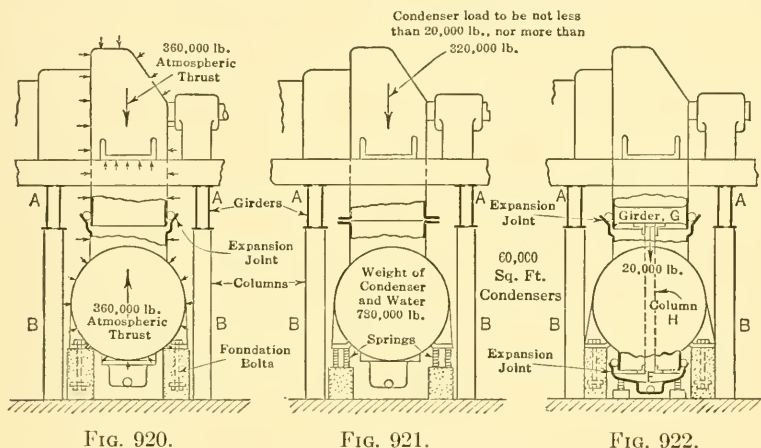


FIG. 920.

FIG. 921.

FIG. 922.

FIG. 920.—Arrangement with Single Unbalanced Expansion Joint between Turbine and Condenser.

FIG. 921.—Arrangement with Direct-Coupled Spring-Supported Condenser.

FIG. 922.—Arrangement with Double Opposed Expansion Joints.

anced downward force (assuming downward discharge to a condenser, as shown in Fig. 920) equal to the difference between the atmospheric and exhaust pressures distributed over the exhaust area; and a similar upward force tends to lift the condenser. If unopposed, these forces would cause the two parts to move accordion-like towards each other. These unbalanced forces are of relatively great magnitude and must be carefully provided for in designing the parts affected, especially the turbine shell, supporting girders (A), columns (B), and footings. The 360,000-lb. forces shown in Fig. 920 are those for a 60,000 sq. ft. condenser with rectangular exhaust opening 10 ft. by 18 ft. 6 in.

(2) **Using direct attachment** of condenser inlet to turbine exhaust flange and suspending the condenser freely from the latter eliminates

the expansion joint and the accompanying unbalanced forces of the arrangement just considered, but it subjects the turbine to the weight of the condenser and its contents, or to such part of this weight as is not supported by other means. Usually the shell of the turbine is designed to be loaded at its exhaust end within certain minimum and maximum limits, hence any additional weight of the condenser and contents is flexibly supported either by means of springs, as in Fig. 921, or by counterweights, as in Fig. 919.

(3) Using double opposed expansion joints in the manner shown in Fig. 922 relieves the turbine and its supports of all condenser loads except a relatively small one. With this arrangement¹⁷ the lower expansion joint F is in effect a balance piston which has an area equal to that of the exhaust opening and is rigidly connected to girders G by means of internal columns H . As the condenser is supported independently, the only load on the turbine exhaust connection is merely that imposed by the weights of the columns H and the lower head F , less that part which may be carried by springs. A design of this character

made possible the installation of a new 75,000-kw. turbine-generator on the same columns that formerly supported a 35,000-kw. unit.¹⁸

(4) Suspending only the condenser shell from the turbine, in combination with independently supported and flexibly connected water boxes, reduces the load on the condenser exhaust flange to merely the weight of the condenser shell.

536. Heat Transmission in Surface Condensers.—(a) The general principles of heat transmission were discussed in Chapter XXVI and are, of course, applicable to the transfers that take

place in surface condensers. Special cognizance, however, should be given to the following:

(1) The transmission of heat between the steam and water in a surface condenser takes place with the steam remaining substantially at constant temperature (t_s), and with the water increasing in temperature in its passage through the apparatus; hence the heat-transfer conditions approximate those shown in Fig. 923 and therefore fall under Case I discussed on page 286, and the mean temperature difference (θ_m) approximates the logarithmic mean value given by Eq. (433), page 290, or by the chart on page 291.

(2) Since the heat surrendered by the steam equals that trans-

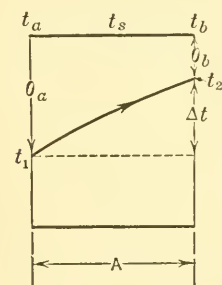


FIG. 923.

¹⁷ Patented by the Worthington Pump and Machinery Corporation.

¹⁸ Proc. N.E.L.A., 1930, pp. 932 and 936.

mitted through the tube walls and also equals that absorbed by the condensing water, it follows that

$$w_s(\Delta h) = \frac{AU\Delta t}{\log_e(\theta_a/\theta_b)} = w_w c_w \Delta t, \quad . \quad . \quad . \quad (642)$$

where A = amount of condenser heating surface, in sq. ft.

c_w = specific heat of the condensing water.

U = thermal transmittance, in B.t.u. per hr. per deg. Fahr. per sq. ft.

Δh = difference in the heat content of the steam entering the condenser and that of the condensate leaving it, in B.t.u. per lb.

$\Delta t = t_2 - t_1 = \theta_a - \theta_b$ (Fig. 923) = increase in the temperature of the condensing water, in deg. Fahr.

w_s and w_w = weights of steam and condensing water, respectively, in lb. per hr.

= Napierian base of logarithms = 2.71828.

Most of the general problems connected with heat transmission in condensers can be solved by using the appropriate pairs of equalities contained in this equation. For *example*: Given the weight of steam (w_s) to be condensed per hour, the amount of heat (Δh) to be abstracted per pound, the temperature of the exhaust steam (t_s), the temperature of the entering water (t_1) and that of the leaving water (t_2), and the heat transmittance (U), then the required extent of tube surface (A) is obtained from the first pair of equalities; and the weight of condensing water (w_w) needed per hour, or the water to steam ratio, w_w/w_s , can be derived from the first and last equalities.

(3) From Eq. (642) it follows that

$$\log_e(\theta_a/\theta_b) = AU/w_w c_w = \frac{U}{w_w/A}, \quad . \quad . \quad . \quad (643)$$

in which (w_w/A) is the number of pounds of circulating water used per hour per square foot of condensing surface, if c_w is taken as unity, a permissible assumption for fresh water except when extreme accuracy is important.

From the last equation it is apparent that, for a given condenser, the ratios θ_a/θ_b and $\theta_b/\Delta t$ remain constant provided no change occurs in U and w_w/A . Thus, if during operation an alteration takes place in these ratios (or in θ_a , θ_b , or Δt , while t_1 remains constant), it indicates a change in some factor affecting U , the cause of which is usually readily determinable. The value of U is largely affected by the water velocity, which depends upon the pump operation, resistances to flow, and the water supply; U is also influenced by a number of other factors which will be considered later in this section. An example of the variation of condenser temperatures with seasonal and other changes is given by the log shown in Fig. 924.

(4) It is also often convenient to note that

$$t_s = t_1 + \Delta t + \theta_b = t_1 + \frac{\Delta h}{(w_w/w_s)} + \theta_b \quad . \quad . \quad . \quad (644a)$$

$$= t_1 + \frac{\Delta h}{(w_w/w_s)} \left[\frac{\theta_b}{\Delta t} + 1 \right] \quad . \quad . \quad . \quad . \quad . \quad (644b)$$

$$= t_1 + \Delta t \div \left[1 - (\epsilon) \frac{-U'}{w_w/A} \right] \quad . \quad . \quad . \quad . \quad . \quad (644c)$$

These equations¹⁹ are useful in showing how the steam temperature, and the corresponding absolute pressure, are influenced by variations in each of the individual factors and ratios appearing in the right-hand members. Typical illustrations are furnished by the curves in Fig. 924.

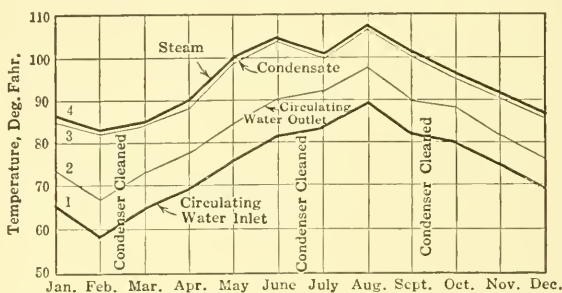


FIG. 924.—Annual Log of Condenser Temperatures (Based on Monthly Averages).

Curves 1 and 2 diverge or converge with changes in (a) the circulating water supply rate, or (b) the turbine steam rate, or (c) the combination of (a) and (b). Curves 2 and 4 diverge when fouling occurs and converge after cleaning. If curves 3 and 4 are not parallel, a change in the proportion of air is indicated.²⁰

If a performance guarantee of a condenser is given on the basis of the heat transmittance (U) for a specified rate of flow of the condensing water, the corresponding exhaust temperature and pressure can then be determined by means of the last equation.

(b) **The overall coefficient, U ,** for a given condenser or tube under given operating conditions can be readily determined by test and the use of Eq. (642). Also, the probable value of U for a tube under

¹⁹ Eq. (644c) is derived from Eq. (643) after rewriting it in the form

$$(\epsilon) \frac{U'}{w_w/A} = (t_s - t_1) \div (t_s - t_1 - \Delta t).$$

²⁰ For further discussion of these curves, see "Condenser Maintenance and Its Influence on Turbine Performance," in *Power*, Aug. 26, 1930, p. 349; also see "Measuring the Temperature Rise of Cooling Water," by D. Park Woolley, *Power Plant Engineering*, July 15, 1932, p. 560.

assumed conditions can be computed by the method discussed in Sect. 338, page 282, provided the needed component heat-transfer coefficients or resistances are known. These components depend upon the respective kinds and conditions of heat transmission involved. In a surface condenser the heat transferred is transmitted by the steam, air, and condensate to the outer surface of the tubes by convection, is caused to flow by conduction through the tube walls and deposits thereon, and is removed by convection from the inner surface by the flowing water. The **film coefficient**²¹ of heat transfer for air-free steam condensing on clean tube surfaces is around 3000 B.t.u. per hr. per deg. fahr. per sq. ft.; and, as explained in Sect. 336(g), page 272, the value of this coefficient is materially decreased by the presence of non-condensable gases, but is generally around 2000. The thermal conductivity of brass tubes at condenser temperature is, from Fig. 479, page 254, about 760 B.t.u. per hr. per deg. fahr. per sq. ft. per in., and the value for Admiralty metal is about the same. The water-film coefficient of heat transfer can be found from Eq. (414), page 268, the principal variable controlling this coefficient being the velocity of the water through the tubes. If the steam is relatively free from non-condensable gases and the tubes are clean, the principal resistance to the heat flow is this water film. With the individual resistances to heat flow known, the numerical value of U can be readily calculated; and since the surface of the condenser is commonly measured on the steam side, Eq. (429b), page 283, should be employed for the usual case in which the water flows through the tube and the steam is on the outside.

Example of the calculation of U for a clean tube follows: Assume that steam at a temperature of 126 deg. fahr. and nearly air-free is to be condensed on the outside surface of a clean brass tube (1.00 in O.D. and 0.902 in I.D.) by water having a temperature of 54 deg. fahr. and a velocity of flow inside the tube²² of 4.84 ft. per sec. Assuming that the ratio of tube length to inside diameter is 72, and that the mean temperature of the water film is 80 deg. fahr., and applying Eq. (414), page 268, it is found that the film coefficient (h_1) on the water side is 980 B.t.u. per hr. per deg. fahr. per sq. ft. of inner tube surface. Then, using Eq. (429b) with this value of h_1 , and with $h_2 = 3000$ and $k = 760$, the value of U is calculated to be 654 B.t.u. per hr. per deg. fahr. per sq. ft. of outer tube surface, which checks closely with Orrok's observed value of 652 obtained under similar conditions. Further, using the three resistances, it is found by solving simultaneous equations that the temperatures of the outer and inner tube surfaces in this case are 110 and 107 deg. fahr., respectively. Hence the mean temperature of the water film is about $0.5 (54 + 107) = 80.5$ deg. fahr., which justifies the original choice.

²¹ "Heat Transmission," by W. H. McAdams, p. 266.

²² These data are taken to correspond to run No. 600 in Orrok's tests of condenser tubes, Trans. A.S.M.E., vol. 32, 1910, p. 1172.

Although the rate of heat transfer in a single clean tube can be predicted closely by a rational analysis, the average for the entire *group* of tubes in a condenser is more difficult to evaluate because of the many factors influencing the transmission. Some of these factors will now be considered.

(c) **The principal factors influencing heat transmission in condensers** are: (1) the proportion of non-condensable gases in the steam and the effectiveness of the air-removal pump; (2) the diameter, thickness, and material of the tubes; (3) the character of the tube surfaces and of the deposits thereon; (4) the effectiveness of the steam distribution throughout the tube nest; (5) the velocity of the condensing water within the tubes; (6) the temperature of this water; and (7) the loading of the condenser, or pounds of steam condensed per hour per square foot of tube surface. In general, the following statements hold: The proportion of non-condensable gas in the condenser should be maintained as low as is feasible, but it is always an uncertain factor. The material and thickness of the tubes have an influence so small as ordinarily to be negligible, but tubes of small diameter are more effective per square foot of surface than those of large diameter, though more difficult to clean. Smooth surfaces are more effective than rough ones; on rusty or etched vertical plates, the condensed steam forms a continuous heat-resisting film of water over the entire surface, whereas on polished chromium-plated surfaces the condensate forms in droplets and the coefficients of heat transfer are *from four to eight times* those with film condensation²³—probably similar relationships hold also for rough and polished condenser tubes. Thick films of oil or grease deposited on the condensing surface, such as those which may form when the steam comes from an internally lubricated reciprocating engine, greatly retard the heat transmission. Such deposits do not occur with steam from turbines. The thermal resistance of a surface film depends in each case on the thickness and character of the material forming the coating. Film thicknesses can be kept within the desired limits by means of various methods of cleaning. Fig. 925 shows in a general way how different degrees of tube fouling affected the transmittance U in one instance. The effect on U of the water velocity (v) through the same tubes is also shown in the same figure. With the usual water velocities it is found that U varies about as v^n . With the new tubes n is around 0.6, and it becomes less with fouling; under ordinary service conditions, the value of n is about 0.5. With cold condensing water the value of U is less than with warm water; never-

²³ McAdams' "Heat Transmission," p. 256.

the-less the colder water is more effective because of the greater temperature difference obtained with it.

Non-uniform distribution of steam velocities, condensate, and occluded gas throughout the tube nest causes the various tubes to be unequally effective, and the coefficient of heat transfer on the steam side is not necessarily constant throughout the length of any given tube. An example of the variation in the heat transfer rates with two differently located tubes of a certain 70,000-sq. ft., two-pass, radial-flow condenser is shown in Fig. 926. Tube *A* was located in the direct path of the entering steam on the outer row at the top of the second pass; tube *B* was near the air outlet at the center of the con-

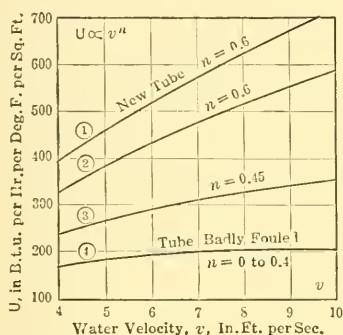


FIG. 925.—Curves Showing the Effect of Various Degrees of Fouling and of Water Velocity on U .

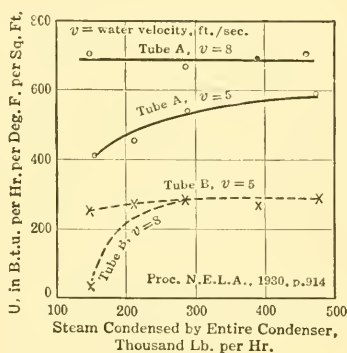


FIG. 926.—Effect on U of Tube Location and Loading.

denser in the first pass; and both tubes were cleaned by sand blasting before the test. With a water velocity of 5 ft. per sec., the heat transfer rate for tube *A* was about twice that for tube *B*; and with a velocity of 8 ft. per sec., the ratio was still greater. This figure also shows the rate of heat transfer with a velocity of 5 ft./sec. as diminishing slightly in both tubes as the total steam condensed per hour is decreased. With a velocity of 8 ft. per sec., tube *B* became air blanketed at the lower rates of condensation, and at 150,000 lb. per hr., the value of U for it was only 15 B.t.u. per hr. per deg. fahr. per sq. ft., which is about the rate of heat transfer from air to water. The effect on U of both the load and the degree of fouling is clearly indicated in Fig. 927 for one set of conditions.

The effect of the condensing water temperature on the absolute pressure obtainable under different loadings on a condenser is clearly illustrated by Fig. 928, Condenser 7 being the one shown in Fig. 912.

Because of the many uncertainties under which a condenser may be operated, especially those as to loading, water velocity, or water temperature, and tube cleanliness, there has been no satisfactory basis heretofore for drawing up condenser guarantees or for checking performances by actual test, and practice in these matters has varied widely.

However, recently there has come into use a somewhat uniform procedure, which is based upon standardized coefficients and correction factors, and which satisfies at least the ordinary commercial requirements.

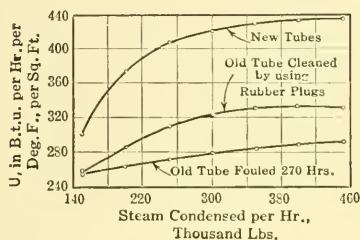


FIG. 927.—Effect on U of Loading and Various Degrees of Fouling.

per sq. ft. per deg. logarithmic mean temperature difference, possible of attainment in well-designed condensers *with commercially clean tubes* (or new tubes) are given by curves ²⁴ in Fig. 929; curves *A* are for three sizes of tubes used with various water velocities, the temperature of the entering water being 70 deg. fahr. and steam being condensed at the rate of 8 lb. per sq. ft. of external tube surface per hr. **Correction factors** for making adjustments for other *water temperatures and loadings* are given, respectively, by curves *B* and *C*.

The respective curves show that U increases about as the 0.5 power of the water velocity through the tubes;²⁵ that it is greater with high water temperatures than low ones; that it has higher values for tubes of

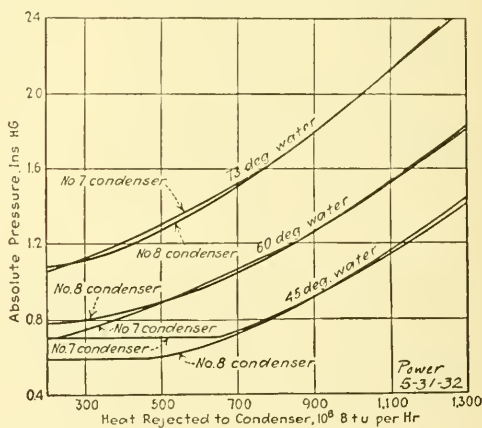


FIG. 928.—Absolute Pressures Guaranteed with 70 Per Cent Clean Condenser.

²⁴ These curves were first published in *Power*, September, 1932, p. 133, with the statement that they had been the basis for many installations made by the following companies: Allis-Chalmers Mfg. Co., C. H. Wheeler Mfg. Co., Elliott Co., Foster Wheeler Corp., Westinghouse Elect. & Mfg. Co., and Worthington Pump and Machinery Corp.

²⁵ As is apparent from the slope of the straight lines obtained when curves *A* are replotted on logarithmic cross-section paper.

small diameter than with large ones, owing to the greater ratio of wetted perimeter to the cross-sectional area; and that it decreases when the loading is reduced below a rate of condensation of 8 lb. of steam per sq. ft. per hr. of external surface. Under the reduced loads, although all the tube surface is available, only part of it is used effectively; hence, under such loads the mean transfer rate with respect to the total surface is diminished. To make allowance for the effect of tube fouling another correction factor, known as the "cleanliness factor," is employed.

(b) The **cleanliness factor** may be defined in general, and especially for the purpose of *design*, as the ratio of the heat transmittance (U) of foul tubes to the transmittance of new tubes subjected to the same operating conditions. The resistance of a tube film depends on the character and thickness of the deposit, which in turn depend upon the degree and character of the foulness of the water used, and the length of time during which the film is allowed to form. The thickness of the film can be reduced by various methods of cleaning, but the cleaning process adds to the operating expense. Under each set of conditions there is a definite interval of time between cleanings that will give an economic balance between the cost of cleaning and the benefits derived therefrom. The average long-time cleanliness factor which may be used for the purposes of design is a matter of considerable conjecture unless the necessary exact data as to the formation and characteristics of the films to be encountered are available.

After a condenser has been placed in operation, the **actual cleanliness factor** can be determined at any time by rather exact experimental methods which have recently been devised,²⁶ and which may be speci-

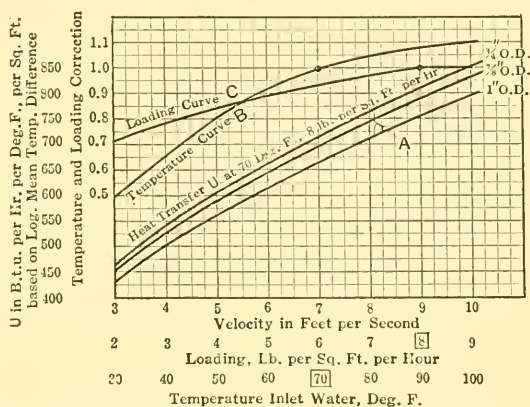


FIG. 929.—Commercial Heat Transfer Rates and Correction Factors for Clean Condensers Used in Connection with Turbines.

²⁶ See "A Test Method for Determining the Quantitative Effect of Tube Fouling on Condenser Performance," by P. H. Hardie and W. S. Cooper, a paper presented before the Metropolitan Section of the A.S.M.E. on Nov. 28, 1932; also, thesis on

fied in the forthcoming revised edition of the A.S.M.E. Test Code for Condensers. The cleanliness factors thus obtained are valuable for reducing to a comparable basis the results of condenser tests made with different degrees of tube fouling, and for correcting acceptance test data to correspond with the tube conditions upon which performance guarantees are based. To accord with the preferred test-methods, the **cleanliness factor** may be **redefined** as the ratio of the average rate of transmittance obtained with one or more representative groups of the regularly installed ("old") tubes within the tube bundle to that obtained with one or more suitably located new tubes installed in the condenser immediately prior to the test and operated under exactly the same conditions.

Note that the *water velocity* should be stated when giving the numerical value of a cleanliness factor. The transmittance with new tubes varies at a higher exponent than that with old tubes; hence the cleanliness factor will vary with the power of the velocity equal to the velocity exponent for the old tubes minus that for the new ones. Since the exponent thus obtained is negative, it indicates that the cleanliness factor decreases with increasing velocity. If test runs are performed at two or more water velocities, this exponent can be determined from the slope of the straight line obtained by plotting the data on logarithmic paper. Unless the velocities differ but slightly, the cleanliness factors should be corrected for inequalities in the velocity before comparing them. *Variation in loading* appears to have little effect on the cleanliness factor; hence its influence can ordinarily be neglected, especially if the variation is slight.

(c) **The preferred method of determining the cleanliness factor** by test is apparently (and very briefly) as follows: Several groups of tubes in various representative positions in the condenser's tube bank are tested in place, each group containing one or more old tubes and one new tube. In a single-pass condenser, the water flowing through the selected tubes is isolated by means of rubber hose jumpers which are carried across both water boxes and are connected to the outside. Each tube is supplied with water from the same source as that for the entire condenser and with a velocity of flow regulated to equal the estimated average for all the other tubes in the tube bank. The data obtained include the temperature of the enveloping vapor, the inlet and outlet water temperatures, and weight of water flowing per hour through each tube. Then, the extent of surface per tube being known,

the same subject presented to Cornell University in June, 1932, by W. S. Cooper towards securing the M.M.E. degree.

the value of U for each tube can be obtained by means of the last two equalities in Eq. (642). The average cleanliness factor for the entire condenser is the ratio of the average transmittance of all the old test tubes to the average of the new tubes. With a two-pass condenser the outlet end of each test tube in the first pass is connected by hose to the inlet end of the corresponding test tube in the second pass. Such tests for cleanliness factor may be conducted simultaneously with the main

condenser test. For further information as to test procedure consult the references already given.

(d) **Standardized values of friction losses encountered in the flow**

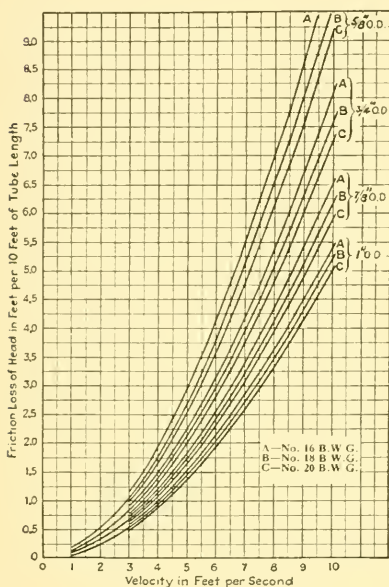


FIG. 930.—Curves for Determining Friction in Condenser Tubes.

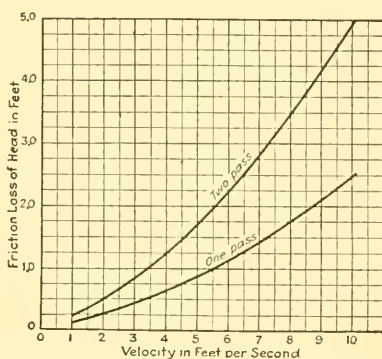


FIG. 931.—Curves for Determining Total Water-Box Losses in Single-Pass and Two-Pass Condensers.

of the condensing water through the tubes and water boxes of condensers have also been agreed upon by the same group of manufacturers to which reference was made in (a) of this section. These friction losses are shown, respectively, in Figs. 930 and 931, which are self-explanatory; these losses, with the unbalanced external heads encountered, are used in determining the size of the motor or prime mover required for driving the circulating pump, and are also important factors to be considered in making economic analyses, such as will be considered in the next section.²⁷

²⁷ For quite complete test data on the friction losses through the various parts of a circulating water system with different water velocities, see Proc. N.E.L.A., 1927, p. 847.

538. Economical Condenser Size, Arrangement, and Operation.—

(a) The required extent of tube surface may be computed from Eq. (642) when the necessary data are given, and which may be based on arbitrary assumptions. The needed tube surface having been determined on such basis and a suitable water velocity through the tubes having been selected, the required number and length of tubes of assumed diameter can then be computed, after which the power needed for pumping the circulating water against the probable heads can be found. But before such computations can be undertaken it is necessary to obtain or assume the requisite primary data. The values to use for U and the friction heads through the condenser can be based upon the discussion and data given in the preceding section. In practice the water-to-steam ratio usually ranges from 40 to 100 or more; the water velocities range from 4 to 10 ft. per sec., but usually are from 6 to 8; and the tubes are $\frac{3}{4}$ in., $\frac{7}{8}$ in., or 1 in. O.D. The results obtained from the computations made in the foregoing manner can be checked by comparing them with the values that have been commonly employed in modern installations.

(b) **Recent condenser practice** in the United States is exemplified by the data given in Table LXXIV which are based upon the N.E.L.A. Report on Condensing Equipment issued in February, 1933. It will be

TABLE LXXIV

CONDENSER DESIGN RATIOS

(As reported by N.E.L.A. for installations made in 1931 and 1932.)

Sizes of units, kw.....	Above 50,000		30,000 to 50,000		Below 30,000	
Number of condenser passes. . . .	Single	Multi	Single	Multi	Single	Multi
Number of units reported.....	6	1	6	1	6	21
Surface, sq. ft. per kw.*.....	0.67	0.76	0.97	0.97	0.83	1.35
Circ. water, G.P.M. per kw.* . . .	1.16	1.11	1.81	1.14	1.63	1.53
Circ. Water, G.P.M. per sq. ft.*..	1.74	1.45	1.87	1.18	1.97	1.13

* Weighted averages for all units reported. (In all cases except one the multi-pass condensers had two passes.)

observed from this table that, in general, single-pass condensers have less surface per kilowatt of turbine-generator but use more circulating water than multi-pass condensers, and that small units use proportionately more condensing surface and water than large ones. It is also of interest to note that prior to 1924 the heat rejected to the condenser

in B.t.u. per kw-hr. of gross generation was from 10,800 to 10,000, for 20,000-kw units and larger, whereas now it is about 7000 in the largest units. The decrease is due partly to improvements in the engine efficiencies of modern turbines, but mainly to the use of higher initial steam pressures and the regenerative cycle. In effect, regenerative feedwater heaters replace condenser surface which would otherwise be required.²⁸

(c) A condenser whose determination is based on arbitrary assumptions, however, even though it conforms to usual practice, will not necessarily be the most economical one to install and operate. The best condenser to use in each case may be determined by an economic analysis based upon a consideration of (1) the operating conditions, (2) the physical factors and limitations, and (3) the annual costs involved. In general, increasing the size of the condenser decreases the annual expenditure for turbine steam but adds to the cost of the condenser; pumping more water through a given condenser lowers the turbine exhaust pressure and steam consumption but increases the yearly pumping cost; and the greater the number of hours during which the equipment is used per year the larger is the warranted initial investment for the equipment. Obviously there is a certain combination of condenser size, vacuum, and rate of water supply, that will be the most economical one to use with each set of conditions. But before the economic values can be determined by analysis it is first necessary to have knowledge of the operating factors to be encountered; and the analysis should not be limited to the condenser alone, but should include consideration also of any related equipment whose size and operation are affected by the condenser. Usually, however, only the influence of the condenser on the steam consumption of the prime mover and on the power required for driving the circulating pump need be considered in this respect.

(d) The **principal operating factors** needed for making an economic analysis for *steady* load conditions are usually established rather easily; but when the load varies throughout the year, and especially when the ultimate future demands are problematical, these factors are matters of considerable conjecture—and then extreme refinements in making the analysis are not warranted. In general, in order to simplify the work, the analysis may be based on *representative or weighted-average values* of the temperature (t_1) of the entering circulating water, and of the weight (w_s) of steam to be condensed per hour, the selection being such that the improved performance obtained during the times when

²⁸ As shown in "Steam-Turbine-Plant Practice in the United States," by V. E. Alden and W. H. Balke, in Trans. A.S.M.E., 1933, F.S.P-55-3a, and in Engineering (London), Aug. 5, 1932.

lower values of these quantities prevail will balance the poorer results occurring when these quantities are higher. The value of t_1 may be obtained from a probable log of water temperatures, like that in Fig. 924. The value of w_s may be derived from a study of the probable weights of steam that will go to the condenser during each hour throughout the year; but it is most readily approximated by using the probable average monthly load curves (which show the magnitude and hours of operation at each load during an average day in each month of the year), and a graph giving the turbine *exhaust rates* under various loadings. (Note that the exhaust rates are not necessarily the same as the turbine steam rates, because some steam may be bled.)

(e) The **most economical condenser** may be **defined** as the one which will show the lowest total combined expenditure annually (1) for the *operation* of the turbine and condenser (this consists primarily of the cost of the turbine steam and of the energy consumed in pumping the condensing water) and (2) for the *fixed charges* (i. e., charges for interest, insurance, taxes, and yearly depreciation or amortization) based upon the initial investment for the condensing equipment. Usually the maintenance costs and other expenses not included above are the same with each of the various condensers that might be used, and then these items are omitted in making comparisons. In making an analysis to determine the economical condenser size and operating conditions, any one of a number of different methods may be followed. Only one method will be considered here, and that very briefly and only in general outline.²⁹ Assuming that the necessary data are available, including a guess at the probable best exhaust pressure to use, and taking a trial value for the diameter and the length of tubes, computations are made to determine the annual costs resulting from the use of various sizes of condensers, each with several different water velocities. Finally, the annual costs are plotted with respect to water velocities and condenser sizes in such manner as to obtain curves from which the economical size and condition of operation can be obtained. The method of computation is given in more detail in the following example, and the curves are discussed in (g) of this section.

(f) **Example:** Given $t_1 = 70$ deg. fahr.; trial exhaust pressure $y = 2$ in. Hg abs.; $w_s = 24,800$ lb. per hr.; $\Delta h = 940$ B.t.u. per lb.; tubes $\frac{3}{4}$ in. O.D., 18 B.W.G., and 16 ft. long; values of U based upon Fig. 929 and a cleanliness factor of 0.70; cost of steam = 35 cts. per 1000 lb.; cost of energy for circulating pump = \$50 per

²⁹ For other methods see: "The Economics of Surface Condenser Selection," by J. H. Smith, in *Power*, May 10, 1932, p. 691; and "Determining the Sizes of Surface Condensers and Circulating Water Systems," by R. P. Moore, in *Steam Plant Engineering*, October, 1932, p. 17.

kw-year; pumping head = 5 ft. plus the tube and water box resistances (Figs. 930 and 931); cost of condenser without tubes = \$3.50 per sq. ft.; fixed charge on cost of condenser without tubes = 15 per cent; weight of each condenser tube = 6.45 lb.; cost of tube material = 25 cts. per lb.; interest on tube cost = 8 per cent; estimated life of tubes = 8 years; operation during 3500 hr. per year; 1 in. change in exhaust pressure alters the turbine exhaust rate 5 per cent. Take as trial sizes of condensers, 2000, 2500, 3000, 4000, and 5000 sq. ft., and with each condenser use trial velocities of 4, 5, 6, 7, and 8 ft. per sec. Proceed as follows, assuming Δh as constant: For each combination of condenser size and water velocity determine the number of tubes, w_s , and³⁰ Δt ; using Eq. (644b) get t_s , from which follows the corresponding exhaust

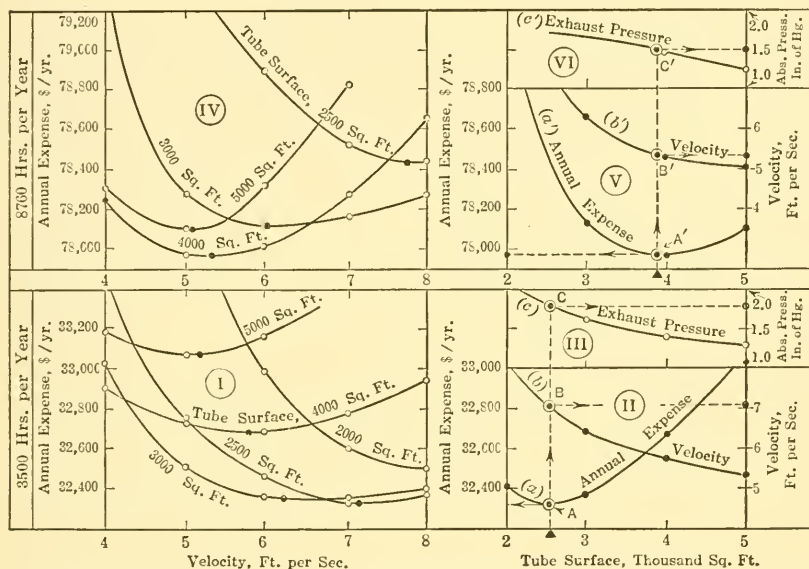


FIG. 932.—Example of Curves for Determining the Economical Condenser Size and Operating Conditions.

pressure y' ; having the value of w_s for the trial exhaust pressure $y = 2$ in. Hg, get the steam rate w'_s corresponding to y' ; determine the combined annual expense for steam, pumping, fixed charges on condenser without tubes, interest on tube cost plus this cost divided by the life of the tubes in years; plot curves like those in the lower part of Fig. 932, and from them determine the economical condenser size, water velocity, and exhaust pressure for the conditions given or assumed.

(g) Curves showing the variation of the annual expense with water velocity are given by (I) in Fig. 932 for the various condenser sizes and conditions considered in the foregoing example; and on each curve is shown the lowest point, which determines the minimum annual expense

³⁰ Since $\Delta t = \Delta h w'_s \div w_w c_w$, in which w'_s is yet to be found and is dependent on Δt , a cut-and-try method must be followed.

obtainable with the corresponding condenser, and also the water velocity with which this optimum condition is accomplished. Plotting these minimum costs and the corresponding velocities and exhaust pressures with respect to the condenser sizes determines curves (a), (b), and (c) shown by (II) and (III) in the figure. From points *A*, *B*, and *C* on these curves it is seen that the condenser should have 2550 sq. ft. of tube surface, the water velocity should be 7.1 ft. per sec., and the exhaust pressure should be 1.8 in. Hg abs, for the given conditions, which include an operating time throughout the year of 3500 hr.

If the condenser unit is operated continuously throughout the year (8760 hr.), other conditions being the same, the corresponding curves would be those shown in the same figure by (IV), (V), and (VI), from which it is seen that with this longer operating period it is desirable to use a larger condenser (3900 sq. ft.), a lower exhaust pressure (1.5 in. Hg abs.), and a lower water velocity (5.4 ft. per sec.) than with the shorter operating time per year.

In this first analysis a two-pass condenser arrangement was assumed. A similar analysis should also be made to determine if a one-pass arrangement would not be still more economical. The tubes were assumed to be $\frac{3}{4}$ in. O.D. and 16 ft. long; it may be desirable to try other diameters and lengths.

539. Condenser Auxiliaries—General.—Regardless of the type of water-cooled condenser used, provision must be made for delivering the condensing water to the apparatus, and also for removing therefrom this water, the condensate, and the air. Sometimes separate pumps are used for handling each of these fluids independently, sometimes two of the fluids are discharged in common by the same pump, and sometimes no pumps whatever are required, as with low-vacuum barometric condensers into which the injection water usually flows without pumping, and from which the mixture of condensing water, condensate, and air is delivered by means of the barometric tube, the supply and removal processes being entirely automatic after operation has once been established. From low-level jet condensers the condensate and condensing water are discharged in common by the *removal pump*, and the air is handled separately by some type of *dry air pump*; also, unless the water supply is at a sufficiently high altitude, an *injection-water pump* is required for this, as with other types of direct-contact condensers. With surface condensers, the condensing water is supplied by a *circulating water pump*, usually assisted by a siphonic action, and the discharge of this water is by gravity; the air and condensate may be removed together by means of a *wet vacuum pump* in installations intended only for fairly low pressures; but when the best results are

desired, the air and condensate are handled separately—the first by “dry-air pumps” and the second by *condensate pumps*. Each of these pumps will be discussed separately in a succeeding section.

540. Condensing-water Supply Pumps.—(a) Injection and circulating water pumps in large installations are always of either the single-stage centrifugal type or the axial-flow propeller type; but with small condensers, pumps of the reciprocating type are sometimes used, as in the unit shown in Fig. 904. A typical small centrifugal pump for this service is shown in section in Figs. 901 and 902; large centrifugal circulating pumps differ in design but slightly from small ones, except as to modifications incident to the greater size and to better opportunities for introducing refinements. One design of a vertical axial-flow circulating pump is shown in Fig. 933; such a pump occupies very little floor space, and, as it is always submerged, no special provision is required for priming it.

(b) If the amount of heat abstracted from each pound of steam condensed is 950 B.t.u., a common value, then the **amount of circulating water** to be handled by a condensing-water pump, *in gallons per minute* (g.p.m.), is

$$G = 1.9w_s/\Delta t = w_u/500 \quad . \quad . \quad . \quad . \quad . \quad (645)$$

in which Δt = rise in temperature in the condensing water, in deg. fahr.

w_s = lb. of steam condensed per hr.

w_u = lb. of condensing water used per hr. (assuming $c_p = 1$).

and $500 = 8\frac{1}{3}$ lb. per gal. $\times$ 60 min. per hr.

If the condenser receives 8 lb. of exhaust steam per kw-hr. and Δt is 11 deg. fahr., then it follows from Eq. (645) that the amount of circulating water required in *gallons per minute* is nearly equal numerically to one and a third times the *output of the turbine in kilowatts*—a simple relation useful as a *rough check* or in making quick, though approximate, estimates.

(c) The **pumping head** against which an *injection pump* of a direct-contact condenser normally works is only that necessary to supplement the lift due to the vacuum; in starting, however, the full static, velocity, and friction heads may be encountered. The *circulating pump* of a surface condenser works against the static head measured from suction to discharge level, the velocity head, and the friction heads at entrance and through the piping, water boxes, and tubes (see Figs. 930 and 931 for the latter two); but after operation is initiated the total head is reduced by the siphonic action due to the use of a comparatively long

vertical discharge pipe with submerged outlet. Usually the full siphonic effect is not available, however, because of the presence of air in the discharge pipe, especially if this air collects in pockets therein. For best operation, the suction lift is limited to 12 ft. with centrifugal condensing-water pump, and as their total pumping heads are low, such pumps always have but a single stage.

(d) The **efficiencies** of centrifugal pumps are usually from 50 to 75 per cent when operated at the capacities, speeds, and heads for which they are designed; however, some large circulating pumps of special design have reached efficiencies from 80 to 87 per cent, and large axial-flow pumps have attained efficiencies up to 86 per cent.³¹

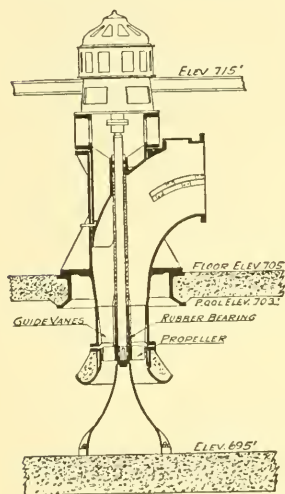


FIG. 933.—Vertical Propeller Circulating Pump.

(N.E.L.A. Pub. 250, February, 1933, p. 20).

(e) The **characteristic** curves showing the relations between the head, speed, power, efficiency, and capacity of a centrifugal pump depend as to form mainly on the curvature and angles of the impeller blades and on the form of the volute or arrangement of the diffusion vanes. The curves are interpreted in much the same manner as those for fans (see Sect. 516).

(f) The **effect of changes in the rotative speed (N) and of impeller diameter (D)**, respectively, are about as follows: *with given diameter* of impeller, (1) the quantity of water delivered varies as N , (2) the head as N^2 , and (3) the power as N^3 ; and *with given speed*, (a) the quantity varies as D , (b) the head as D^2 , and (c) the power as D^3 . These relations are approximate, but accurate enough for many purposes.

(g) The **average energy consumption of circulating pumps** and rates of water supply as reported by the N.E.L.A.³² in 1929 for seven two-pass condensers and nine single-pass condensers, were as follows:

³¹ For discussions of axial-flow pumps see: "Axial-Flow Pump for Condensing Water," in *Steam Plant Engineering*, September, 1932, p. 23; "Power and Mechanical Engineering Exhibition, New York," in *Engineering* (London), Jan. 13, 1933, p. 37; "Flexible Performance at High Efficiency Characterizes Adjustable-Blade, Propeller-Type Pumps," by R. E. P. Sharp, in *Power*, May, 1933, p. 242; and N.E.L.A. Publication 250 on *Condensing Equipment* (Feb., 1933), pp. 20 and 30.

³² See Proc. N.E.L.A., 1929, p. 1464.

Kind of surface condenser	Two-Pass	Single-Pass
Power consumption, kw. per 100 sq. ft.....	7.2	6.9
Energy consumption, kw-hr. per million B.t.u. of heat transferred.....	1.24	0.94
Pounds of water per hr. per sq. ft.....	590	796
Pounds of water per 1000 B.t.u. transferred.....	102	111
Velocity of water in tubes, ft. per sec.....	6.2	5.7

(h) Condensing-water pumps are classified as **essential auxiliaries**, along with condensate pumps. Certainty of operation is of primary importance with such apparatus, but considerations of simplicity, initial cost, maintenance expense, and economy of energy consumption, though secondary, should also be carefully investigated. When a shutdown of the unit served would involve serious consequences, *twin pumps* are often installed, or a single pump may be provided with *dual drives*, both being electric or one electric and the other steam, as in Fig. 899. When twin pumps are selected, each is usually of a size to meet half of the full requirements; and when they serve a divided condenser, a cross connection between their discharge pipes is usually included, together with suitable valves, so that either pump can serve either half of the condenser in an emergency, or one pump can be shut down during light loads or when the water is very cold. The present trend, however, appears to be towards maintaining constant rates of water supply, except when very light loads coincide with water temperatures below 40 deg. fahr.

(i) The **drives** for circulating pumps may be either electric or steam; and, as already noted, energy can be saved by using variable speeds. Steam drives are particularly suitable for flexible operation, and often the exhaust steam from them can be employed to advantage in the energy balance of the station. These pumps, however, are quite commonly driven by one-speed or two-speed induction motors; and sometimes dual pumps and dual drives of unequal capacities are used so as to afford flexible combinations to meet varying conditions. In general, the steadier the load, the less is the need for adjustable-speed drives; and a plant not having reliable electrical interconnections must have some of the essential auxiliaries steam-driven, both for emergency use and for starting up the plant.³³

³³ See "Auxiliary Drive for Steam Power Stations," by F. H. Hollister; "Steam Driven Auxiliaries for Power Plants," by W. P. Dryer (in *Electrical Engineering*, January, 1932); and "Electrically Driven Auxiliaries for Steam Power Stations"

541. Condensate Pumps; Removal Pumps.—(a) Pumps for removing the condensate from condensers, in addition to the other heads encountered in the suction and delivery systems, work against heads corresponding to the vacuum in the condenser. They are commonly of the centrifugal type, and have but a single stage when the total head is comparatively low (e.g., 150 ft.), and two stages for higher heads. A single-stage pump of this kind is shown in Fig. 901; one with two stages, in Fig. 934. The shaft glands are commonly sealed with condensate under pressure to prevent air leakage; and, as the water handled is warm and under reduced pressure, it should be supplied to the pump under at least 3 ft. *positive head* (preferably 5 or 6 ft.) to prevent vapor binding and cavitation from occurring within the pump. Condensate pumps, being “essential equipment,” are often installed in duplicate or have duplicate drives; much of the discussion regarding drives for

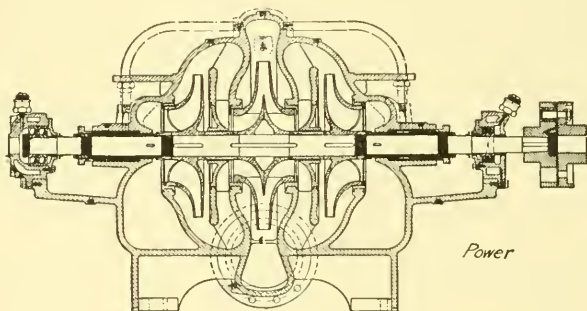


FIG. 934.—Example of a Large Two-Stage Condensate Pump.

(One of two used with the condenser shown in Fig. 912. The shaft packing is sealed by water from the second stage.) From *Power*, May 31, 1932, p. 826.

condensing-water pumps applies also to them. The efficiency of a well-designed two-stage condensate pump may reach 65 per cent under favorable conditions. The capacity of such pumps should be somewhat in excess of normal requirements (e.g., 25 to 50 per cent) so as to be able to meet emergency conditions, as when the condensate rate is increased because of poor turbine performance, overloading, cessation of bleeding, or leakage of circulating water through tube joints or ruptures. Condensate pumps are usually designed to force the condensate through at

(with bibliography), by L. W. Smith, all presented before the A.I.E.E. at the January, 1932, meeting; “Steam vs. Electric Drive,” in *Power*, Feb. 9, 1932, p. 195; also, “Selecting the Motor to Suit the Pump,” by E. C. Dieffenbach, in *Power*, April 28, 1931, p. 656, and “Choice of Type of Motor,” by H. M. French, in *Southern Power Journal*, December, 1932, p. 34.

least one low-pressure heater into an elevated surge tank or to a combined deaerator, extraction heater, and surge tank. The static head is constant, but frictional resistances increase with the rate of flow. A float in the hotwell can be arranged to control the pump speed, if variable drive is used.

(b) **Removal pumps** used with jet condensers handle the warm mixture of condensate and injection water and, like the condensate pump, work against the atmospheric pressure in addition to the other heads. They are ordinarily of the single-stage centrifugal type. See Figs. 898 and 899.

542. Air Removal—General.—(a) The amount of air entering a condenser per unit of time depends (1) on air infiltration into the part of the system which is under vacuum (i.e., leakage inward through the low-pressure stages and glands of the turbine and through joints and porous walls of the exhaust piping and condenser shell), (2) on the mechanically entrained air brought into the system in the boiler feed-water, and (3), in jet condensers only, on the air brought in by the injection water. Raw water may contain 4 or 5 per cent of air by volume at atmospheric pressure and temperature, but the proportion may be less than 1 per cent if the water comes from an open heater and still less if it has been subjected to special deaeration, as in the deaerator. The amount of air being handled by a given condenser can be readily determined by experimental methods, but the amount to be removed from a prospective condenser is quite problematical as it depends on the design, construction, and care of the system. It was estimated, in 1919, that for a first-class *surface-condenser system*, the air leakage with respect to condensing capacity should not be higher than that shown in Table

TABLE LXXV

ESTIMATED AIR LEAKAGE WITH SURFACE CONDENSERS

With tight system and good operation.*

Steam to Condenser, lb./hr.	Estimated Air Leakage, lb./hr.	Parts by Weight per 100,000	Steam to Condenser, lb./hr.	Estimated Air Leakage, lb./hr.	Parts by Weight per 100,000
100,000	28	28.0	400,000	55	13.9
200,000	46	23.0	500,000	55	11.0
300,000	53	17.7	600,000	55	9.2

* Based on estimates contained in "Air Pumps for Condensing Equipment," by Frank R. Wheeler, in Mech. Eng., Dec., 1919, p. 926.

LXXV, the leakage rate being relatively less with large condensers than with small ones. Modern condenser systems show even better results. A number of central-station condensers ranging in size from 20,000 to 70,000 sq. ft. and tested in 1929 showed air leakage in nearly all cases as falling between 2 and 6 cu. ft. per min., the average being 5 and the maximum 15, size, however, not being a determining factor.³⁴ A capacity of 5 or 6 cu. ft. of air per minute is often provided in dry-vacuum air pumps with *surface condensers* regardless of condenser size. With *jet condensers* the air handled is that brought in both by the steam and by the injection water, and may be at least double the amounts given in Table LXXV for surface condensers. In many modern plants the outlets of the removal apparatus are equipped with air meters or indicators, usually inverted bell meters or orifice meters, so that at any time the amount of leakage can be easily checked, at least approximately. But an air pump has to handle not only the dry air but also the vapor with which it is fully saturated.

(b) The **ratio of vapor to dry air** by weight (w_v/w_a) in a saturated mixture of air and water vapor may be found for *any* temperature and pressure by the equation

$$\frac{w_v}{w_a} = \frac{\bar{V}_a}{\bar{V}_v} = \frac{R_a T / P_a}{\bar{V}_v} = \frac{0.7536 T}{p''_a \bar{V}_v} = \frac{0.7536 T}{(p''_m - p''_v) \bar{V}_v}, \quad (646a)$$

where the subscripts a , v , and m denote air, vapor, and mixture, respectively; $\bar{V}$ = specific volume in cu. ft. per lb.; $R_a = 53.3$; T = abs. temp., deg. fahr.; P = abs. press. in lb. per sq. ft.; and p'' = abs. press. in in. Hg.

When the pressure of the mixture is less than 4 in. Hg abs., a simpler and more convenient equation that involves only a ratio of pressures may be derived.³⁵ This equation is

$$\frac{w_v}{w_a} = 0.622 \frac{p''_v}{p''_m - p''_v} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (646b)$$

(c) The **weight of vapor per pound of dry air** in a saturated mixture under various condenser pressures and temperatures may be found from Eq. (646) or from Fig. 935. It is interesting to note that at a given partial pressure and temperature the density of air is greater than that of water vapor, the density ratio being

$$\rho_{\text{air}} / \rho_{\text{steam}} = R_{\text{steam}} / R_{\text{air}} = 85.7 / 53.3 = 1.6.$$

³⁴ Proc. of N.E.L.A., 1929, p. 1464.

³⁵ See page 996.

The **air-handling capacity** which the air-removal apparatus for a condenser must have, and the **work** to be done by such apparatus, depend on (1) the rate of air leakage into the system, and (2) the specific volume of the air-vapor mixture under the pressure and temperature existing at the pump inlet. For example, the volume of dry air handled at 70 deg. fahr. is 399 cu. ft. per lb. if received at 1 in. Hg pressure; and it is 199.5 cu. ft. if the pressure is 2 in. Hg. The effect of suction temperature on the volumes of dry and saturated air and on the relative weights of the two to be handled at a given pressure are shown in Table LXXVI.

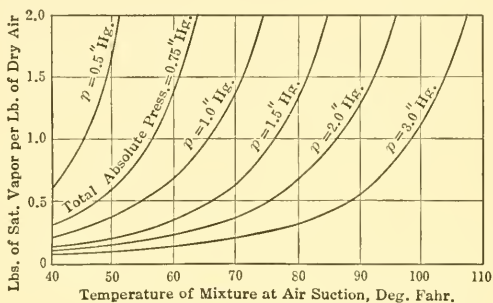


FIG. 935.—Weight of Vapor per Pound of Dry Air at Various Condenser Pressures and Temperatures.

TABLE LXXVI

EFFECT OF SUCTION TEMPERATURE ON REQUIRED AIR PUMP CAPACITY, WITH
SUCTION PRESSURE = 1 IN. HG

Suction temperature, deg. fahr.....	70	60	50
Specific vol. of <i>dry</i> air, cu. ft. per lb.....	399	392	384
Specific vol. of saturated air, cu. ft. per lb.....	1530	820	602
Ratio (weight of saturated mixture)/(weight of dry air) ..	2.75	1.67	1.35

(d) The **relative capacities needed** and the **amounts of work done** by any type of air-removal apparatus having isentropic compression are shown in Fig. 936 for two comparative cases, in both of which the pressure in the condenser is 1 in. Hg abs. and the temperature is 70 deg. fahr., but one diagram is for dry air and the other is for air initially saturated with water vapor. The difference between the areas representing the net work is very noticeable and clearly shows the effect of the vapor present, and also emphasizes the desirability of condensing as much of the vapor as is expedient before handling the mixture. The advantage of using intercoolers (or condensers) between the stages of two-stage arrangements of air-removal apparatus may also be easily visualized from these diagrams.

543. Air-Removal Apparatus.—(a) The various kinds of air-removal devices for condensers may be classified broadly as either **dry vacuum** or **wet vacuum**. The first kind handles only the air and vapor with which it is saturated, and includes steam-jet air ejectors, hydraulic air ejectors, and positive displacement air pumps of the reciprocating

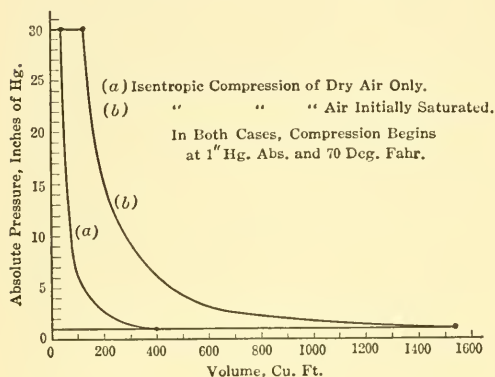


FIG. 936.

piston and rotative types. The second kind handles both this mixture and the condensate simultaneously.

Air-removal apparatus of all kinds are in effect air compressors which receive the saturated air from the condenser and compress it to or above the atmospheric pressure against which it is discharged.³⁶

(b) If the compression process is a polytropic one with an exponent n , then, from Eq. (88), Part I (page 96), the *volume of the mixture at the end of compression* is

$$V_2 = V_1 (p_1/p_2)^{1/n}, \quad . \quad . \quad . \quad (647)$$

in which V_1 is the initial volume, and p_1 and p_2 are the initial and final absolute pressures.

(c) The **net work done on the mixture by the pump** is, from Eq. (134), Part I (page 130),

$$\text{Work} = \frac{n}{n-1} P_1 V_1 \left[(p_2/p_1)^{\frac{n-1}{n}} - 1 \right]. \quad . \quad . \quad . \quad (648)$$

The *actual compression process* is intermediate between the isothermal and the isentropic for the reciprocating pump; but, in general, this process depends on the type of pump employed and upon the degree of cooling effected during the process. With mechanical compressors this cooling is accomplished by means of water injection, water jacketing, or, in the case of multi-stage compression, by intercooling, perhaps combined with the other expedients. With steam-jet air ejectors the air-vapor mixture comes in direct contact with the live steam, and heat is thereby transferred *from* this steam to the mixture while in the ejector.

³⁶ The principles of air compressors discussed in Chapter X, Part I, are applicable to such apparatus.

The chief form of cooling employed during compression with the ejector is that accomplished by an intercondenser, which is clearly applicable only to the multi-stage type; but in some cases a water-cooled jacket is also placed around the diffuser, or compression tube, of the ejector.

(d) The **steam-jet air ejector** is the air-removal apparatus most commonly used in present-day practice; this is because of its simplicity, diminutive size, small first cost, and low maintenance and operating expense. These ejectors may be either of the *single-* or *multi-jet type*, some applications of which are shown in Figs. 898, 903, 906, and 938.

The steam jet, or jets, entrain the "air" in a mixing chamber, and force it through a throat and a diverging compression tube, commonly called a *diffuser*, against a higher pressure than that at the air inlet. The air-entraining capacity per pound of steam used is greatly increased by employing a number of small nozzles instead of a single large one discharging steam at the same rate. A **single-stage ejector** has one set of nozzles and one compression tube and is suitable only for moderate compression ratios, say up to 8 to 1 as a maximum, because its use beyond this ratio involves very high rates of steam consumption. A **two-stage ejector** has a second set of nozzles which furnish steam to compress the mixture from the first stage and force it through a second compression tube up to the final delivery pressure. For very low pressures (e.g., below 1 in. Hg abs.) three stages may be used.

Both the single- and multi-stage ejectors may be designed to operate either **non-condensing** or **condensing**. If a condenser is placed between two stages of an ejector, it is called an **intermediate condenser**, an **interstage condenser** (or intercooler); also the last stage often delivers into a condenser, called the **after condenser**; and when both are used the apparatus is of the **inter-and-after condenser** type. Such condensers can be either of the *direct-contact type* as in Fig. 937 (also 896), or of the single-pass or multi-pass *surface type*, as in Fig. 938, the inter-and-after condensers being located in the same shell. A two-stage

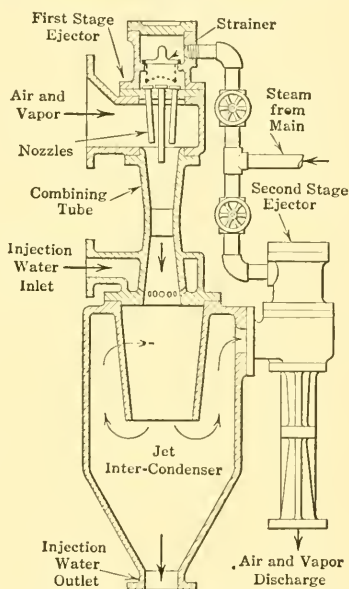


FIG. 937.—Steam-Jet Air Ejector with Jet Inter-Condenser.

inter-condensing type of ejector uses only about 40 per cent of the amount of steam required by a two-stage non-condensing type. The condensing water used by an inter-and-after *surface* condenser can be raw water, but preferably it is condensate taken from the main hotwell and then used as feedwater; thus the energy supplied by the steam used to operate the ejector is quite fully recovered. The intermediate condenser is often provided with a separate section through which cold

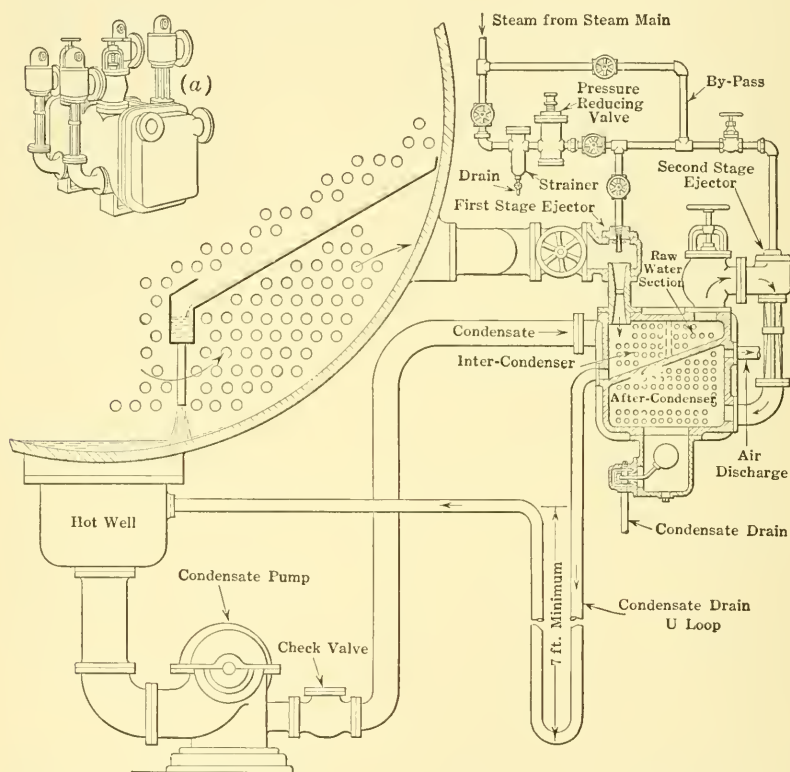


FIG. 938.—Steam-Jet Air Ejector with Inter- and After-Condensers of the Surface Type.

raw water is passed whenever there is not sufficient condensate or make-up water to condense all the steam from the first stage; this is especially useful in starting the main condenser.

The condensed steam from the first stage, in which the discharge pressure is considerably below atmospheric, is usually drained to the main condenser hotwell through a water-sealed loop having an effective height of 7 ft. or more, as in Fig. 938 (also 903), to prevent the

return of air to the main condenser; but this condensate is sometimes removed by means of a trap or pump, or by drainage through a loop to some other chamber in which the pressure is lower. If the recovery of condensate from the after surface condenser is also desired, a suitable trap or pump must be used. Two or three sets of ejectors (as in (a) in Fig. 938), sometimes of unequal capacities, are commonly installed so as to provide for flexibility and reliability of operation, the same inter-and-after condenser being used for all. An outside source of condensing water should be provided for starting. When the supply of main condensate is insufficient, as during light-load periods, it may be supplemented by returning to the hotwell of the main condenser either some of the condensate just used in the ejector or water from the surge tank, or raw water may be substituted.

Each steam-jet ejector must be designed for the specific conditions under which it is to operate, and it will not function satisfactorily, if at all, under other conditions. If the steam pressure is variable, a reducing valve or orifice should be used to maintain a constant pressure below the lowest to be encountered in the steam main. In general, the ejector is very reliable, but it will not operate satisfactorily if there is (1) insufficient cooling water, (2) low steam pressure, (3) excessive back pressure, (4) scale or dirt in the nozzles, or (5) faulty drainage so that water floods the inter- or after-condenser.

Because of the great number of variables involved, equations to express the performance of steam-jet air ejectors are as yet incomplete. The success of commercial applications has depended largely on developments resulting from experimental investigations, but unfortunately not many of the results of these investigations have been published.³⁷ Apparently, the most suitable compression ratio per stage is 5 or 6 to 1, and the ratio of steam to dry air, by weight, with two-stage condensing ejectors should not be over 9 or 10 to 1 when the condenser pressure is 1 in. Hg. Compared with a two-stage condensing-type of ejector, one of the same type and capacity but with three stages uses about 30 per cent less steam at 2 in. Hg, 33 per cent less at 1 in., and 63 per cent less at 0.5 in.³⁸

Single-stage, high-capacity, steam-jet air ejectors are also used for priming the water pumps and then are known as **priming ejectors**;

³⁷ See "The Production of a Vacuum in an Air Tank by Means of a Steam Jet," by F. R. B. Watson, paper before (British) Inst. of M.E., Feb. 17, 1933, reprinted in (London) Engineering, Feb. 23, 1933, p. 230, and March 3, 1933, p. 262 (discussion, p. 255); "Fluid Jets and Their Practical Applications," by Mollenby, in (British) Inst. of Chem. Engineers, Oct. 26, 1928.

³⁸ "Three-Stage Air Ejector Uses Less Steam," by W. D. R. Morgan, in Power, July 27, 1926, p. 127, and discussion, Oct. 5, 1926, p. 530.

also they are employed as evacuators for rapidly reducing the pressure in the steam space of the condenser when quick starting is desired, and to evacuate the condensing water system so as to reduce the starting head on the circulating pumps.

(e) **Hydraulic air pumps** operate on the entrainment principle and are of either the *jet-ejector type* or the *hurling type*. The **hydraulic-jet air ejector** uses single or multiple high-velocity jets of water to entrain the air and discharge it through a compression cone.⁴ The apparatus resembles steam-jet air ejector but is larger for the same capacity. It combines the functions of an air pump and, to some extent, a condenser, the latter increasing the effective head on the water and consequently the water velocity and entrainment. The apparatus is quite simple, but becomes unstable when called upon to handle air in excess of the designed capacity. In the **hurling type**, alternating slugs or layers of air and water are projected through a collector cone in which the slugs of water act as pistons in carrying the intervening laminations of air through it. One example of this type is afforded by the Leblanc air pump, the elements of which are shown in Fig. 939. The hurling water, drawn into the pump chamber by the vacuum, passes through the stationary distributors (*B*), and the rapidly moving vanes in the revolving element (*R*) discharge it in successive sheets which entrain the air entering the collecting cone and force it out through the diffuser. To start the vacuum, live steam may be temporarily admitted at *S*, if necessary. Pumps of this type combine high air-removal capacity with compactness and are able to carry overloads; and because of the increased water velocity under high vacuum the capacity increases with the vacuum. These pumps operate at high speed and are driven either by small steam turbines or by electric motors.

For a given capacity, hydraulic air pumps of all types use much more power than do the piston air pumps, which will be considered next, and are slower in starting; they use relatively large amounts of water (which must be quite clean so as to prevent the clogging and erosion of their parts), and they waste the condensed vapor. The jet type requires separate supply pumps. Often a hydraulic air pump is arranged to discharge into a sealing tank from which the water is returned to the supply pump; thus the same water is used repeatedly, after deaeration in the tank. Warm water should not be used because it increases the vapor pressure in the combining chamber of the air pump.

(f) **Rotative dry vacuum pumps** (R.D.V. pumps) are of the positive-displacement, piston type and are essentially air compressors in which the reciprocating pistons are operated from rotating cranks, thus differing from the crankless direct-acting type. As no "air"

can enter the compression cylinder during the suction stroke until that remaining in the clearance at the end of compression has expanded to a pressure below that at the intake, special means are taken to reduce the quantity of the clearance air or to minimize its effect. The following expedients may be employed, singly or in combination, for the purposes of increasing the capacity and vacuum obtainable and reducing the power requirements with given size of cylinder and piston speed: (1) the clearance volume is made as small as possible; (2) water jackets are provided, especially around the clearance space, so as to make the compression and expansion processes approach the isothermal; (3) the compression is accomplished in two stages, with intercooling; (4) the inlet valve is provided with an auxiliary equalizing passage (as that at *a* in Fig. 940) which, just before the return stroke starts, exhausts the clearance space momentarily into the other end of the cylinder which at the time is at inlet pressure—thus, when the suction stroke starts, the clearance space is nearly at intake pressure; and (5) water may be used in the cylinder. A Worthington-type *two-stage*

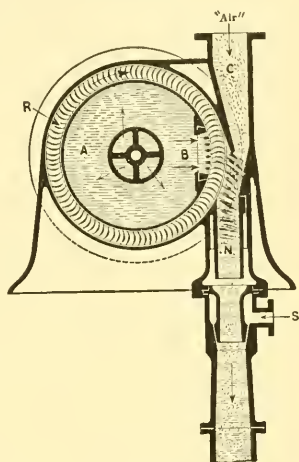


FIG. 939.—Leblanc Air Pump.

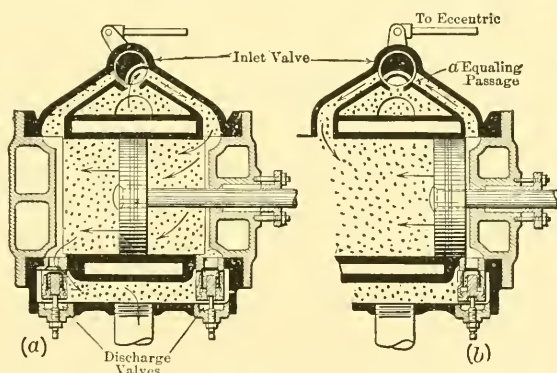


FIG. 940.—Dry Vacuum Pump with Equalizing Passage.

vertical compressor with but a *single* cylinder is shown in Fig. 941. This pump has the advantages of the single-cylinder mechanism while having a two-stage effect. The upper end of the cylinder constitutes the first stage, in which, however, no appreciable compression takes

place; but the effect of the clearance is largely eliminated by this transfer stage, whereby the volumetric efficiency is increased so much that, though single-acting, nearly the same size of pump as the double-acting one can be used for a given capacity. The valves are thin, flexible strips of ribbon steel, similar in appearance to clock spring stock, which seat tightly on ground-faced slotted seats, and in opening flex against solid curved guards, the ends remaining in contact with the

seat at all times. No mechanically actuated valves are used. When the piston descends, the air enters the head end of the cylinder; on the up stroke this air is transferred, at in-

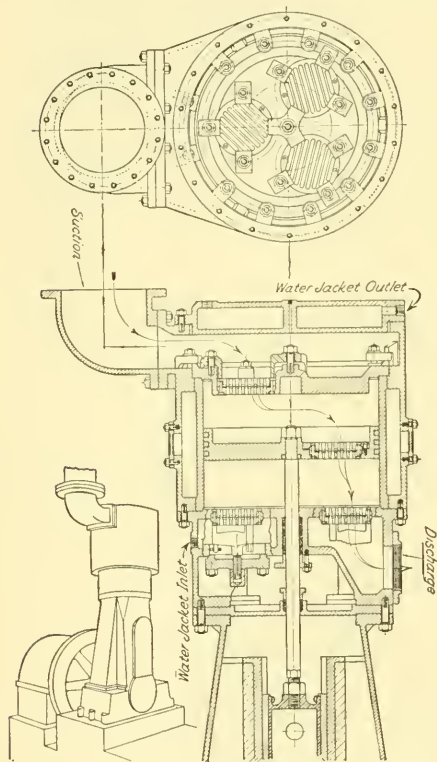


FIG. 941.—Two-Stage Single-Cylinder Dry Vacuum Pump.

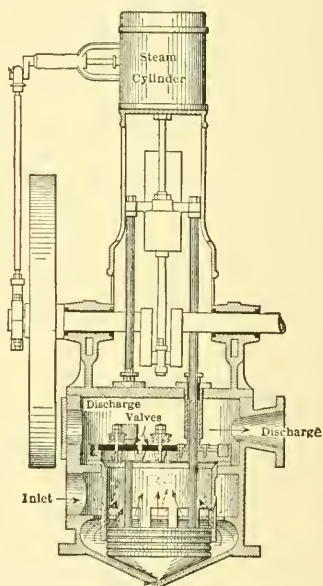


FIG. 942.—Wet Vacuum Pump, Edwards Type.

take pressure, through the valves of the piston to the crank end of the cylinder, wherein during the next stroke it is compressed and delivered through the lower valves. A pump of this kind for a 30,000-kw. turbine is driven by a 30-hp. motor. Other equivalent two-stage arrangements are also used. The use of water within the cylinder of an air pump to overcome partially the effect of clearance is particularly useful in positive-displacement pumps of the rotating piston or lobe types. Although

water is in the cylinder, the pump is still classified as a dry-vacuum one if it discharges only the air.

(g) **Wet-vacuum positive displacement pumps** may have either reciprocating pistons or rotary pistons (or lobes), and they pump both water and the air simultaneously. The reciprocating pumps may be of the crankless direct-acting type, as in Fig. 904, or of the rotative type with crank and flywheel, one example of which is given by the Edwards type of pump shown in Fig. 942. In this pump all valves are dispensed with except those for discharge. The air and water which collect in the base of the pump are displaced and forced into the pump cylinder by the conical end of the plunger as it approaches the bottom of its stroke, then are caught above the plunger when it ascends, and are discharged ultimately through the discharge valves in the upper deck. As in most pumps of this kind, there is a lip around the valves so that they will always be water-sealed to prevent air leakage.

544. Condenser Maintenance.—(a) But little care and expense are ordinarily involved in keeping condensers of the *direct-contact* type and their auxiliaries in good condition; but with *surface* condensers, the cost of tube replacement and the work and expense of maintaining the tubes properly clean and tight may be items of considerable magnitude. With bad condensing water the life of the tubes may be only two or three years even though tubes of extra thickness are used; and even in plants having fairly good circulating water a period of five years has often been the average life, although in some cases tubes have been used for ten years or more. The cost of cleaning and renewing of condenser tubes in a large station may amount to \$100,000 or more per year.

(b) The **cleaning of condenser tubes** must be done at intervals which depend upon the rate at which the dirt, slime, and scale of various kinds deposit on the tubes. The frequency of cleaning may vary from twice a week in one plant to twice a year in another. The economic period between cleanings depends on the character of circulating water, the cost of extra steam required with poor vacuum, and the expense connected with the cleaning, including the outage of the turbine during the process.

One common method of cleaning condenser tubes is by shooting **plugs** through the tubes by compressed air or water under pressure; these plugs are made of rubber, metal, rubber and canvas, or metal and canvas. If the deposit is oily or very slimy the plug method of cleaning is not entirely satisfactory and is therefore supplemented by **sandblasting** or using **wire brushes**, say semi-annually or annually. In some cases a *hard scale* can be removed only by **drilling** or using **acids**. Sometimes the tubes are cleaned by **baking**; this process con-

sists of drawing all the water out of the tubes and then maintaining the steam space full of steam at a temperature of about 185 deg. fahr. for a period of 5 hours. The condenser is then allowed to cool for 30 minutes after which it is flushed out with cold water. The baking process requires about 9000 lb. of steam to clean a 42,700-sq. ft. condenser³⁹ as contrasted with 4 men working 12 hours to shoot plugs through a condenser of this size.

Preventive measures, however, may be used to reduce very greatly the rate of deposit on the tubes. Clogging from débris may be avoided by the use of trash racks and screens. Other deposits may be reduced by treating the water chemically,⁴⁰ as by chlorination.

(c) **Chlorinating the circulating water** before it enters the condenser is a method particularly suitable for preventing (1) the formation of slime on the walls of the tubes and (2) the growth of mussels on the walls of the intake and discharge tunnels for the circulating water. Experience has shown that very small amounts of chlorine, such as 0.5 part per million, injected into the circulating water intermittently, say 4 minutes out of every 90, will be very satisfactory. One company reports⁴¹ that an average gain in vacuum of 0.24 in. Hg during a year has been obtained by chlorination and that a gain of only 0.06 in. will pay for the cost of operation and maintenance of the chlorinating equipment. The chief disadvantage of this method of cleaning is the extra hazard introduced because of the escape of chlorine gas.⁴²

(d) **Condenser tube failures** have been classified by the A.S.M.E. Special Research Committee on Condenser Tubes into the following types:⁴³

I. *General Thinning or General Corrosion*.—The wasting away or thinning down of the tube. The inside surface of the tubes is evenly corroded without the formation of pits or dezincified spots. Little or no basic salt on the tube.

II. *General Dezincification*.—The tube is generally not thinned down to any considerable extent. The zinc is taken out, leaving a mass of spongy and more

³⁹ See "Steam Condenser Practice and Performance," by F. J. Chatel, Mech. Eng., March, 1927, p. 230.

⁴⁰ See "The Treatment of Condenser Cooling Water," by Applebaum and Browne, in Steam-plant Engineering, December, 1932, p. 22.

⁴¹ See "Chlorine Treatment of River Water at Powerton Station," by K. E. Stoll, Power, Feb. 2, 1932, p. 168.

⁴² Additional references: "Chlorinating Circulating Water Keeps Condenser Clean," by V. M. Frost and W. F. Rippe, Power, Dec. 17, 1929, p. 961; Proc. N.E.L.A., 1930, pp. 921-930; Engineering, Aug. 5, 1932, p. 167; "Slime and Mussel Control in Surface Condenser," by Powell, in Combustion, April, 1933, p. 7; "Bibliography of Deterioration of Condensing Equipment" (1845 to 1931), prepared under the auspices of the A.S.M.E.

⁴³ See Mech. Eng., June, 1929, p. 442.

or less brittle copper behind. This shows when broken a red or brick-like fracture extending from the inside partly or completely through the tube wall. It is frequently more or less irregular, so that the tube wall is completely dezincified in places and on other places not entirely so.

III. *Deposit Attack or Pitting*.—Small, round pits or pinholes beneath green salt and cuprite. Foreign deposits often accompany this type of corrosion. When these pits or pinholes contain plugs of copper, this is known as “plug type” dezincification.

IV. *Tailed-Type Pitting*.—Pits formed by the removal of the scale and having a deep section at the upstream side and a shallow tail at the downstream side.

V. *True Erosion*.—The tube is worn away by the action of solids in the circulating water, often accompanied by corrosion.

VI. *Impingement Attack*.—Corrosion caused by air bubbles and generally confined to the inlet end of tube. This may be uniform thinning or it may appear as grooving.

VII. *Scale Thinning*.—Scale kept thin by chemical action, resulting in corrosion and in isolated pits.

VIII. *Solution*.—Scale absence due to the presence of acids, etc., resulting in general thinning.

IX. *Splitting and Season Cracking*.—Splitting in a longitudinal plane caused by internal strain in the tube metal or undetected defects within the tube walls. Season cracking may be due to the same cause as splitting, and in addition may be due to stresses set up in expanding and rolling the tube ends in the tube sheets.

The **frequency** of occurrence of the different types of failures is indicated by Table LXXVII, as obtained by the A.S.M.E. Committee.

TABLE LXXVII *

CONDENSER TUBE FAILURES

As reported by 141 plants, having 2,400,895 tubes installed.

Cause of Failure	Average Yearly Replacements	
	Number	Per Cent of Total Tubes Installed
Corrosion.....	59,255	2.5
End corrosion.....	16,218	0.7
Erosion.....	5,935	0.2
Corrosion and splitting.....	2,710	0.1
Splitting.....	5,320	0.2
Scale.....	584	0.02
Unclassified.....	2,062	0.08
Total.....	92,084	3.80

* See Mech. Eng., February, 1928, p. 171.

The **location of a plant** affects its percentage of the tube failures very materially. For example, in 1928 the A.S.M.E. committee reported that (1) for the Coastal Sections of this country the average yearly percentages of failure were 11.1 for the Metropolitan area, 10.1 for the Pacific Coast, 5.5 for the Gulf Section, and 4.5 for the Atlantic Coast other than the Metropolitan; (2) for the Great Lakes Section the average is only 0.5, although in East Chicago it is 11.65; (3) for the Mississippi Valley Section the average is 4.4 on the West Side and 3.5 on East Side with 5 stations reporting over 10. In the Coastal Sections, 5 stations had yearly replacements greater than 20 per cent and one was more than 30 per cent.

(e) The **prevention of tube failures** is a matter of great importance, and much research work is still being done in this field. However, sufficient information is now available to assist both the condenser designer and operator in reducing the loss from this cause. The following conditions should be fulfilled to the degree that is commercially feasible in any given case: (1) Avoid circulating water containing injurious substances, if possible, or use tubes made of material that is least corroded by the water that must be used; (2) use a satisfactory method of attachment of tube to tube sheet; (3) avoid mechanical injury to the tube during cleaning, tightening of packing, or renewal; (4) avoid excessive vibration of the tubes; (5) keep air out of the tubes; (6) avoid turbulence in the circulating water; and (7) avoid stray electric currents. All circulating water contains air, but the trouble from this source is increased by having excessive turbulence of the water in the intake canal, or by infiltration of air through the pump glands. To prevent an accumulation of air in the water boxes and upper rows of tubes, the water boxes should be properly vented. Severe turbulence of the circulating water at the entrance end of a tube will be caused by failure to have a stream-line entrance. For this reason the slotted ferrule is now avoided in many plants that formerly used it.

(f) **Condenser leakage** consists of air or water, or both, leaking into the steam space. *Air may find its way* into this space through the condenser joints, porous castings, exhaust connections, atmospheric relief valve, turbine shaft seal, and turbine casing. *Water may leak* into the steam space through cracked or porous tubes, around the joints at the ends of the tubes, and through defective tube sheets.

Air leakage is not always easily detected. However, by closing the valve between the condenser and the air pump for a short time, say 10 minutes, the vacuum gage will show a rapid increase in the condenser pressure if there is much air leakage; this assumes that the leakage is extremely large relative to the very small amount of air that ordinarily

enters the condenser with the steam. As soon as it is discovered that there are air leaks their location is next to be determined. If the leaks are large, the deflection of the flame of a lighted candle passed over the suspected joints may give indication; but a better method, generally, is to spray oil of peppermint over the suspected places while another person notes the odor at the discharge of the air pump. Both of these methods have the advantage of being applicable while the condenser is in operation. If the condenser is out of service the steam space may be filled with compressed air, say 5 lb. per sq. in. gage, and the leaks located by means of soapsuds; this is about the only way to locate leakage through a porous casting or around valve stems. In large plants, air meters are now commonly installed in the discharge line from the air pump, and thus the rate of air coming out of the condenser may be approximately determined at any time.

Water leakage is generally meant when the term *condenser leakage* is used. If a condenser has water leaking into the steam space during the steam rate test of a turbine or engine, clearly the test data cannot be considered reliable unless the rate of condenser leakage has been determined. Furthermore, in ordinary operation, leakage of circulating water into the steam space is very undesirable because impurities are thus introduced into the feedwater. Consequently large central stations keep constant watch of the purity of the condensate.

The **methods of determining condenser leakage** recommended by the A.S.M.E. are the following:⁴⁴ (1) qualitative silver nitrate; (2) quantitative silver nitrate; (3) short direct-weight; (4) standard direct-weight; and (5) electrolytic-conductance. Methods (1) and (2) can be applied only with circulating water containing an appreciable amount of salt, as when sea water is used; method (1) is a very sensitive one to *detect* any leakage, but one of the other methods must always be used to find the *rate of leakage*. With fresh circulating water, method (4) is generally preferred. This method consists in measuring the leakage water collected in the hotwell during a period in which no steam is allowed to flow to the condenser while the regular vacuum is maintained as nearly as possible.

When a condenser is not in operation, leaks are commonly found by filling the entire steam space with water and noting whether any water comes out of any tube. When leaky tubes are found they are often plugged at each end with tapered wooden or cork plugs; these defective tubes are replaced at some later time when the condenser may be more conveniently kept out of service.

⁴⁴ For details, see "Power Test Codes, Instruments and Apparatus," Part 21, Chapter I, Condenser Leakage Tests.

(g) The **shock or hydrodynamic method of testing condenser tubes** consists in applying a water hammer to each tube in order to detect the weak ones. This method is used in the U. S. Navy because one of its great advantages lies in the probability of detecting weak tubes *before* failure has actually occurred; hence applying this test while a vessel is in port will greatly reduce the chances of tube failure while at sea. To produce a satisfactory pressure wave for this test, one end of the condenser tube must be *tightly plugged*, as a small leak at the end will greatly reduce the excess pressure produced when the pressure wave reaches the plug. The pressure wave is initiated by allowing a weight to fall a definite height and strike a plunger whose movement is resisted by the hydraulic pressure in a chamber connected by copper tubing and nozzle to the other end of the tube under test.

The application of relatively high pressures for a very minute interval of time, say from 2000 to 3000 lb. per sq. in. for 0.03 or 0.04 sec., as with the shock method, does not have any bad effect on the good tubes but will cause the badly pitted ones to fail. On the other hand, tests have shown that a pressure of 1000 lb. per sq. in. applied *hydrostatically* to condenser tubes will appreciably shorten their life; this is due to the relatively great length of time of the application of the pressure with this method.⁴⁵

⁴⁵ For details of conducting the hydrodynamic test, see "Shock Method of Testing Condenser Tubes," by Lieut. Commander H. A. Seiller, *Power*, May 8, 1928, p. 807.

CHAPTER XLV

FEEDWATER HEATERS, EVAPORATORS, PUMPS, AND TRAPS

545. Introduction.—(a) The **purpose** of a feedwater heater in a steam-generating plant is to transfer heat from steam to the boiler feedwater; that of an **evaporator** is to use steam to vaporize, and thus purify, raw or partly treated make-up water which is later condensed for boiler feed. In order that the feedwater system may function properly in relation to the plant as a whole, certain **pumps** are necessary to supply water to the several pieces of equipment in accordance with the cycle of operation employed, and also to feed the heated and purified water to the boilers. **Traps** or similar devices may be needed for disposing of the condensate formed in certain kinds of feedwater heaters, as well as that produced in the steam piping.

Various other kinds of **heat-recovery apparatus**, certain forms of which are known as **heat exchangers**,¹ are often incorporated into the feedwater system for the purpose of transferring heat to the feedwater from some other fluid which must have its temperature reduced as a necessary part of its own action within the plant. Such devices are used to supplement the feedwater heaters; and they will be taken up at a later point in this chapter after the main types of feedwater heaters, and their applications, have been considered.

(b) The **design** of the apparatus associated with the heating, evaporating, and purifying of water is fairly simple, but the equipment itself is often connected to other elements of the plant in a rather complex arrangement, the purpose of which is to assist in producing the best economic performance of the plant. In general, stations employing high pressures and temperatures operate on the *regenerative cycle*, and in this case the number, type, and disposition of the feedwater heaters used become of considerable importance. The use of make-up evaporators, as well as various other kinds of heat exchangers in the feedwater system, must also be taken into account during the design of

¹ Although the term **heat exchanger** may be applied to any apparatus in which heat is transferred from one substance to another, its use in power-plant practice is usually confined to those pieces of equipment in which both of the substances are *liquids*.

the plant. The principle of the regenerative cycle is given in Part I, Sect. 215; a detailed study of regenerative units and of the performance of stations operating on the regenerative cycle, either alone or in combination with other arrangements, has been made in Sects. 274 and 279, to which attention is directed. Curves of comparative performance with several arrangements of equipment are given in Fig. 349 (page 130); and the ways in which feedwater heaters may be incorporated into various combinations of equipment will be considered later in Chapter XLIX.

Although complex arrangements of feedwater heaters are essential to the proper operation of large central stations, it should not be inferred that the field of usefulness of such equipment is restricted to installations of this type; this apparatus also forms an important part of most other kinds of steam-generating plants, no matter how simple the "hook-up" may be, because of the many benefits attending its use. Evaporators are usually installed wherever a sufficient supply of good make-up water is not available from other sources.

A. FEEDWATER HEATERS

546. Advantages of Feedwater Heaters.—The principal advantage derived from feedwater heating is the improvement in the thermal efficiency of the plant; however, the following additional advantages are also obtained: (1) an increase in the steam-generating capacity of the boilers, since less heat need be transmitted in them to the working substance for each pound of steam produced; (2) a reduction of the strains produced in the boiler metal by the introduction of relatively cold feedwater; and (3) the partial deposition, outside of the boiler, of the scale-forming impurities contained in the water. In plants operating without reheating or extraction, each 12-deg. rise in the temperature of the feedwater will result in a saving of approximately 1 per cent of the fuel, provided the heat used to raise the water temperature would otherwise be wasted. Against the financial gain thus produced, must be set the fixed and operating charges associated with the heater and its allied equipment; however, in almost all cases the final result shows a considerable profit through the use of the heater.

547. Classification of Feedwater Heaters.—(a) On the basis of heat transfer from the steam to the water, feedwater heaters of the more common types may be classified as follows: (1) **open**, or **direct-contact heaters**, in which the steam and water mix; and (2) **closed** (non-contact or surface) **heaters**, in which the steam and water are separated by metallic walls through which heat is transmitted from one medium

to the other. A further classification is based upon the source of the steam used for heating, as follows: (1) **exhaust-steam heaters**, in which the heat is derived from steam exhausted by prime movers or auxiliaries; (2) **extraction heaters**, which make use of steam bled from the intermediate stages of turbines, or from the receivers of multiple-expansion engines; and (3) **live-steam heaters**, which use steam direct from the boiler.

(b) In addition to the two major classifications given above, still others may be used to designate feedwater heaters. Thus, a classification may be based upon the pressure of the entering steam. A heater which receives steam at a pressure less than atmospheric is known as a **vacuum, or primary, heater**; one which receives steam at about atmospheric pressure is known as an **atmospheric, or secondary, heater**; and one which uses steam at a pressure considerably above that of the atmosphere is designated as a **pressure heater**. A **through heater** is one which handles all of the steam exhausted by an engine, but condenses only a fraction of it, the remainder passing on to a condenser; an **induction heater** is one in which all the entering steam is condensed, and in which the quantity of steam admitted depends solely on the inductive action caused by the condensation of the steam previously drawn in. Feedwater heaters often combine the functions of heating and purifying; thus there are **live-steam purifiers**, and **deaerating heaters**; sometimes, also, a feedwater heater is designed as an integral part of a chemical softening system. A **storage heater** is one which provides for a reserve supply of hot water to meet sudden demands. Since all feedwater heaters, regardless of structure or arrangement, perform the same basic function, only the more common types will be considered in detail in the following sections.

548. Open-type Feedwater Heaters.—(a) Heaters of this type are fitted with cascading or spraying devices to break up the water and thus bring it into more intimate contact with the steam. The mixing takes place within a suitably shaped shell which is usually rectangular or circular in cross-section; shells designed for low-pressure service are made of cast iron, whereas those subjected to high internal pressures are made of cast steel, or are fabricated from boiler plate.

(b) One form of **low-pressure open heater**, which is designed to use exhaust steam, is shown in Fig. 943. The shell contains a number of removable pans within which a considerable amount of suspended matter is retained as the feedwater flows downward through the heater, and on the outer surfaces of which some scale is deposited as the water trickles downward in a film over these surfaces and is heated by the steam; thus the water is partially purified as well as heated.

(c) The **live-steam purifier** is a modification of the device just described and differs from it mainly in the use of steam direct from the

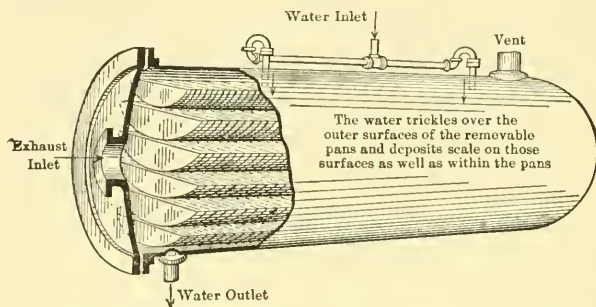


FIG. 943.—One Form of Open Feedwater Heater with Removable Trays.

high-pressure mains. Its chief advantage is due to a greater degree of purification; this is because many of the dissolved solids which are not precipitated at the temperature of the exhaust steam will be thrown out of solution at the higher temperature of live steam.

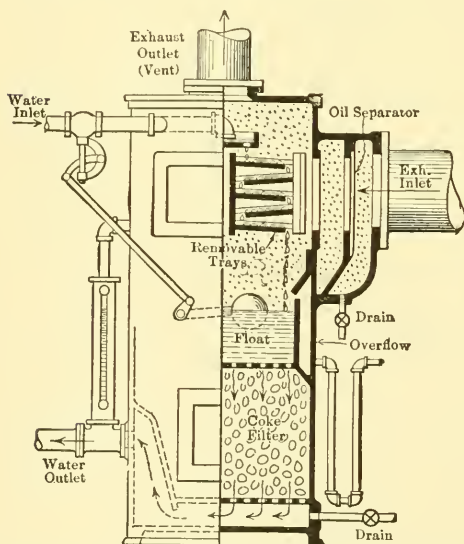


FIG. 944.—A Form of Open Feedwater Heater Containing an Integral Filter and an Oil Separator.

In every open heater the condensed steam becomes a part of the boiler feedwater; hence all precautions must be taken to prevent oil from entering the heater, and thus finding its way into the boiler.

(e) The **deaerating heater** is a special form of open heater which

is arranged to remove, to the greatest possible extent, the dissolved gases in the feedwater. A heater of this type is shown in Fig. 962 (page 985), and its usefulness in improving the properties of the feedwater is brought out in Sect. 576. A deaerating heater should always be *vented* through a condenser (see Fig. 962), the function of which is to liquefy any water vapor which is mixed with the non-condensable gases leaving the heater, and also to recover some of the heat which would otherwise escape from the system with the gases and vapor. In modern practice **vent condensers** are often attached to open heaters of all the types so far described, for the purpose of heat recovery as just explained.²

(f) The **jet type** of open heater produces a thorough mixing of the steam and water in a manner somewhat similar to that of the jet condenser shown in Fig. S98 (p. 882). The chief advantage of this heater is its ability to handle large quantities of feedwater in an apparatus of comparatively small size; but it must usually be designed with several independent groups of nozzles which will allow its capacity to be varied over a wide range. This type has recently come into use for extraction feedwater heating in central stations; however, other types of open heaters may also be used for this service.

549. Closed Feedwater Heaters.—(a) This type of heater consists essentially of a number of straight or curved tubes, connected at their ends to waterboxes, or heads, and enclosed in a shell which is usually cylindrical. The **tubes** are of corrosion-resistant metal, commonly brass or copper; their sizes are given on the basis of outside diameters, and their thicknesses are specified according to the B.W.G. (Birmingham Wire Gage) scale. Standard dimensions and other data for the tubes used in both feedwater heaters and condensers are given in Table LXXVIII. Closed heaters are made in several different forms, three of which are shown in Figs. 945, 946, and 947. The heaters shown in Figs. 945 and 946 are of the type in which the water passes through the tubes and the steam surrounds the tubes, while the design illustrated in Fig. 947 has the opposite arrangement. The first water-tube type (Fig. 945), which is the more common, may have its effectiveness considerably reduced through the formation of scale within the tubes if feedwater of poor quality is handled. Storage heaters are usually of the **steam-tube type** (Fig. 947) since the shell of such a heater serves as a reservoir containing a reserve supply of hot water. If the latter type is fitted with curved tubes it is a simple matter to crack loose any adherent scale by chilling the tubes and causing them to contract suddenly. The coil-tube design shown in Fig. 946 provides

² See Power Plant Engineering, July 15, 1930, p. 724; and May 1, 1931, p. 486.

TABLE LXXVIII

DATA ON STANDARD TUBES FOR CONDENSERS AND FEEDWATER HEATERS

Diameter in Inches		Thick- ness B.W.G.	Weight in Pounds		Surface, Sq. Ft. per Foot of Length		Inside Area, Sq. In.	Water Velocity, Ft. per Sec. for 1 G.P.M.
Outside	Inside		Per Foot	Per 100 Sq. Ft. Outside	Outside	Inside		
1/2	0.232	10	0.5674	433.4	0.1309	0.0608	0.0424	7.56
	0.260	11	0.5280	403.3	0.1309	0.0682	0.0532	6.03
	0.282	12	0.4908	375.0	0.1309	0.0738	0.0625	5.13
	0.310	13	0.4454	340.2	0.1309	0.0812	0.0755	4.25
	0.334	14	0.4010	306.2	0.1309	0.0874	0.0876	3.662
	0.356	15	0.3626	276.9	0.1309	0.0932	0.0995	3.220
	0.370	16	0.3276	250.1	0.1309	0.0969	0.1076	2.980
	0.384	17	0.2978	227.3	0.1309	0.1005	0.1157	2.772
	0.402	18	0.2582	197.2	0.1309	0.1052	0.1269	2.530
	0.430	20	0.1888	144.3	0.1309	0.1125	0.1452	2.209
	0.452	22	0.1400	108.0	0.1309	0.1200	0.1600	1.875
	0.470	24	0.1000	76.8	0.1309	0.1280	0.1760	1.600
	0.488	26	0.0700	54.0	0.1309	0.1360	0.1920	1.375
	0.506	28	0.0400	31.2	0.1309	0.1440	0.2080	1.150
5/8	0.357	10	0.760	464.2	0.16362	0.0937	0.100	3.205
	0.385	11	0.700	428.0	0.16362	0.1010	0.117	2.730
	0.407	12	0.649	396.5	0.16362	0.1070	0.130	2.474
	0.435	13	0.531	365.0	0.16362	0.1142	0.149	2.154
	0.459	14	0.520	317.6	0.16362	0.1205	0.165	1.942
	0.481	15	0.460	281.0	0.16362	0.1260	0.1817	1.766
	0.495	16	0.421	256.7	0.16362	0.1299	0.1925	1.663
	0.509	17	0.380	232.2	0.16362	0.1331	0.2035	1.579
	0.527	18	0.326	199.5	0.16362	0.1382	0.2181	1.472
	0.555	20	0.238	145.5	0.16362	0.1455	0.242	1.328
	0.577	22	0.180	108.0	0.16362	0.1530	0.268	1.150
	0.595	24	0.130	76.8	0.16362	0.1600	0.294	1.000
	0.613	26	0.090	54.0	0.16362	0.1680	0.320	0.875
	0.631	28	0.060	31.2	0.16362	0.1760	0.346	0.750
3/4	0.482	10	0.950	484.0	0.19635	0.1265	0.1822	1.760
	0.510	11	0.870	443.2	0.19635	0.1335	0.2043	1.566
	0.532	12	0.807	411.0	0.19635	0.1400	0.223	1.441
	0.560	13	0.718	365.8	0.19635	0.1470	0.247	1.302
	0.584	14	0.640	326.0	0.19635	0.1530	0.268	1.202
	0.606	15	0.563	287.0	0.19635	0.1590	0.289	1.111
	0.620	16	0.514	262.0	0.19635	0.1627	0.3019	1.069
	0.634	17	0.464	236.1	0.19635	0.1660	0.3157	1.019
	0.652	18	0.396	202.0	0.19635	0.1706	0.3339	0.962
	0.680	20	0.289	147.0	0.19635	0.1780	0.3632	0.883
	0.702	22	0.210	108.0	0.19635	0.1860	0.390	0.775
	0.720	24	0.160	76.8	0.19635	0.1940	0.416	0.675
	0.738	26	0.110	54.0	0.19635	0.2020	0.442	0.588
	0.756	28	0.080	31.2	0.19635	0.2100	0.468	0.500
1	0.607	10	1.153	503.0	0.2291	0.158	0.288	1.114
	0.635	11	1.046	456.0	0.2291	0.167	0.317	1.012
	0.657	12	0.984	429.0	0.2291	0.172	0.339	0.944
	0.685	13	0.862	376.0	0.2291	0.1795	0.367	0.874
	0.709	14	0.763	333.0	0.2291	0.186	0.394	0.815
	0.731	15	0.630	275.2	0.2291	0.1915	0.420	0.764
	0.745	16	0.603	266.0	0.2291	0.1951	0.4360	0.736
	0.759	17	0.548	239.0	0.2291	0.1988	0.4524	0.710
	0.777	18	0.468	204.5	0.2291	0.2034	0.4742	0.676
	0.805	20	0.340	148.4	0.2291	0.2107	0.5090	0.631
	0.827	22	0.260	108.0	0.2291	0.2180	0.5340	0.588
	0.845	24	0.210	76.8	0.2291	0.2260	0.5580	0.500
	0.863	26	0.160	54.0	0.2291	0.2340	0.5820	0.425
	0.881	28	0.110	31.2	0.2291	0.2420	0.6060	0.350
1	0.732	10	1.34	512.0	0.2618	0.192	0.421	0.764
	0.760	11	1.22	466.0	0.2618	0.200	0.455	0.706
	0.782	12	1.12	428.0	0.2618	0.205	0.479	0.671
	0.810	13	0.99	378.0	0.2618	0.213	0.515	0.625
	0.834	14	0.88	336.0	0.2618	0.2183	0.5463	0.588
	0.856	15	0.77	294.2	0.2618	0.225	0.576	0.558
1	0.870	16	0.700	268.0	0.2618	0.2277	0.5945	0.540
	0.884	17	0.632	241.7	0.2618	0.2314	0.6138	0.524
	0.902	18	0.540	206.0	0.2618	0.2362	0.6410	0.503
	0.930	20	0.389	149.0	0.2618	0.2435	0.6793	0.474
	0.952	22	0.310	108.0	0.2618	0.2510	0.7070	0.425

a relatively large amount of heating surface in proportion to the size of the shell. In closed heaters of all forms, the material of the shell is cast iron for low or moderate steam pressures, and cast steel or steel plate for high-pressure operation.

(b) Adequate provision for the **free expansion and contraction** of the tubes is an essential requirement of closed-heater design. This is easily accomplished in curved-tube types, since the tubes are capable of moving relatively to the shell, even though their ends are rigidly fastened into the tube sheets. In the straight-tube forms it is also desirable that the tubes be fastened rigidly into each tube sheet in order that leakage may be prevented; consequently in modern practice the tubes extend from a fixed head which is bolted to the shell, to a **floating head** which is entirely independent of the shell, although supported within it, as shown in Fig. 945.

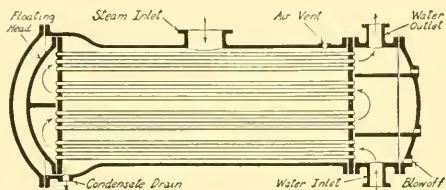


FIG. 945.—Closed Heater of the Multi-Pass, Floating-Head Type.

(c) Closed heaters should preferably operate on the **counterflow** principle, especially when the heater is of the through type and is placed in an exhaust line leading from an engine to a condenser. A closed heater may be **single-pass** (Fig. 947) or **multi-pass** (Fig. 945), depending upon the number of times the

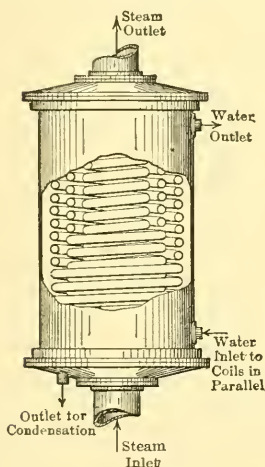


FIG. 946.—Closed Heater of the Coil Type.

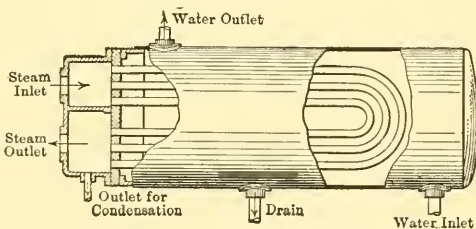


FIG. 947.—Closed Heater of the U-Bend Type.

water traverses the length of the tubes. Generally the greatest number of passes is four, although heaters have been designed with many more passes than this. In an induction heater the steam usually enters the shell and surrounds all the tubes, the resulting condensate being drained

off through a trap; however, a through heater may contain baffles which are so arranged that the *steam* makes several passes through the shell in a direction counter to the flow of the water within the tubes. Heaters are sometimes arranged so that the water flows over the heating surfaces in a thin layer, such devices being known as **film-type** heaters.

(d) The **structural features** of closed heaters resemble those of surface condensers in many respects; this is especially true of the heaters using low-pressure steam. In heaters for *high-pressure service* (e.g., 500 lb. per sq. in. or more), the basic arrangement of the parts remains the same as that of heaters for more moderate pressures (see Fig. 945), but special forms of construction are necessary to increase the strength of the entire structure and secure tight joints; thus in one design hook bolts are used with separate ring flanges for attaching the covers of both the floating head and the shell, these parts thereby being mechanically locked together in a definite relation to each other; in another design the fluid pressure on the head-cover is resisted by segmental shear pieces set into recesses in the extended wall of the head, no bolts being used for the purpose of withstanding the pressure.³

550. Energy Equations for Feedwater Heaters.—(a) With one exception, which will be considered later, the basic relation governing the **transfer of heat** from the steam to the water in any feedwater heater may be expressed by the following equation, which presupposes steady flow, as well as no heat losses through the shell:

$$w_s (h_a - h_b) = w_w (h_2 - h_1), \quad . \quad . \quad . \quad . \quad (649)$$

in which w_s and w_w are, respectively, the quantities of steam and of feedwater flowing through the heater, in lb. per hr.; h_a and h_b are, respectively, the heat contents of the "hot" fluid entering and leaving, in B.t.u. per lb.; and h_2 and h_1 are, respectively, the heat contents of the feedwater leaving and entering, also in B.t.u. per lb.

Since the condensate formed in an **open heater** becomes a part of the feedwater, the two liquids reach a common temperature as they leave the heater; and the quantities h_b and h_2 , from Eq. (649), are therefore equal, in this type of heater.

For moderate feedwater temperatures, the specific heat (c_p) of water is close to unity; hence, for such cases, the term $(t_2 - t_1)$ may be substituted for the term $(h_2 - h_1)$ in Eq. (649), the terms t_2 and t_1 being, respectively, the temperatures of the feedwater leaving and entering the heater. An examination of Fig. 948 will show that the error introduced by this substitution is less than 2 per cent for temperatures up to

³ See Heating, Piping, and Air Conditioning, May, 1933, p. 248.

265 deg. fahr., but increases rapidly as the temperature is raised above that point; hence consideration should be given to the magnitude of the error produced by considering $c_p = 1$ in any given case.

(b) When an open heater is **vented** to a condenser, which is the exception referred to in (a), the basic energy equation should include an item to indicate the heat content of the vapor-gas mixture withdrawn through the vent. For this particular case the arrangement of the heater is shown diagrammatically in Fig. 949, and the corresponding equation, assuming steady flow and no heat losses through the shell, is as follows:

$$w_s h_a + w_w h_1 = w_f h_2 + w_3 h_3. \quad . \quad . \quad . \quad . \quad (650)$$

In this equation, w_f is the weight of the feedwater, and w_3 is the combined weight of the vapor and non-condensable gases leaving the heater,

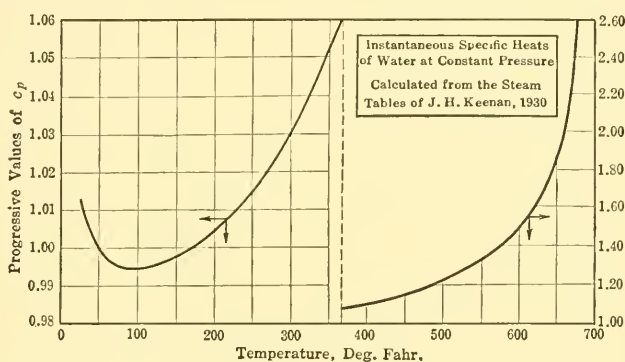


FIG. 948.—Instantaneous or Progressive Values of Specific Heat, c_p , of Water.

each in lb. per hr.; and h_3 is the heat content of the vapor-gas mixture, in B.t.u. per lb. of mixture. The remaining symbols are the same as those used in Eq. (649).

The heat content of the weight of vapor-gas mixture vented per hour is given by the following equation:

$$w_3 h_3 = w_v h_v + w_g c_p (t_g - 32), \quad . \quad . \quad . \quad . \quad (651)$$

where w_v and w_g are, respectively, the weight of the vapor and of the non-condensable gases leaving the heater, in lb. per hr.; h_v is the heat content of the vapor, in B.t.u. per lb., and t_g is the temperature of the gases leaving through the vent, in deg. fahr. The specific heat, c_p , of the non-condensable gases may usually be taken as 0.24, since air is the predominating gas and its temperature is not very high.⁴

⁴ See Plate VIII in the Appendix.

If the water vapor at the exit from the vent is in the form of either superheated or saturated steam at a pressure less than 2 lb. per sq. in. abs., and at a temperature (t_x) between 200 and 600 deg. fahr., its heat content, as given on page 423, is

$$h_x = 1057 + 0.46t_x. \quad . \quad . \quad . \quad . \quad . \quad (652)$$

The properties of mixtures of air and water vapor are considered in detail in Sect. 581, page 995.

551. Heat Transfer in Closed Feedwater Heaters.—(a) The rate of heat transfer in closed heaters is affected by the same factors as those that determine the rate in surface condensers; in fact, *every* feedwater heater, with the exception of the through type, is really a condenser.

In every closed heater a **terminal temperature difference** exists between the entering steam and the feedwater leaving the heater; this difference may amount to 8 or 10 deg. when saturated steam is used in a properly proportioned heater which is well maintained,⁵ and to 20 deg. or more if the design is poor or the operating conditions are unfavorable. When superheated steam is used in closed heaters, especially in connection with the regenerative cycle, the feedwater may be raised to a temperature higher than the saturation temperature corresponding to the absolute pressure of the steam, but not equal to that of the entering superheated steam.

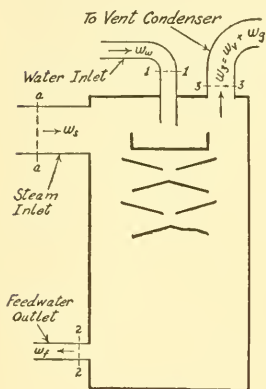


FIG. 949.

(b) The most important factors which affect the transfer of heat in closed heaters are (1) the velocity and viscosity of the water, (2) the degree of cleanliness of the inner and outer tube-surfaces, (3) the condition of the steam, (4) the presence of non-condensable gases in contact with the tubes, and (5) the material of the tubes. The influence of these factors on heater performance is similar to their effect on condenser operation, and has been considered in detail in Sects. 336(c), 338, and 536. In heaters, the surface on the water side of the tubes is more easily kept clean than it is in condensers, since the feedwater is usually of much higher quality than the cooling water used in condensers; however, under certain conditions scale will form from the precipitation of solids as the water is heated, and the tubes will thus become fouled. The surface on the steam side of the tubes may have its effectiveness enormously reduced by a thin film of

⁵ See Power Plant Engineering, January, 1933, p. 26.

oil from the exhaust steam used for heating; also, a stagnant film of air or condensed steam on this surface will produce the same result, hence closed heaters are sometimes vented through an orifice to the main condenser, or the design is such that the non-condensable gases are removed through the drain trap with the condensate.⁶

(c) The effect of **increased water velocity** upon the coefficient of heat transfer in closed heaters is shown approximately by the curves in Fig. 950, which represents the composite results of heater tests reported in recent years by the Prime Movers Committee of the N.E.L.A.⁷ Although increased velocity causes a greater rate of heat transmission and a consequent reduction in the amount of heating surface required for a given capacity, it also entails larger friction losses through the tubes and waterboxes, with a resulting increase in the power required to pump the water. Therefore in designing a heater system these factors must be taken into account so that the arrangement which involves the lowest overall cost may be obtained.⁸ Heaters are commonly designed so that the water velocity through the tubes will be from 4 to 6 ft. per sec. during normal operation; and the pressure drop between the entrance and exit connections varies considerably with the design, the range being from 4 to 12 lb. per sq. in.

(d) The **film-type** heater provides a means for obtaining very effective heat transmission without the use of excessively high velocities. In the water-tube form of this type the feedwater is usually required to flow in a helical path through an annular space between the outer heat-transfer tube and an empty inner tube, each particle of water thus coming more frequently into contact with the heating surface than is the case when plain tubes are used. With water velocities only slightly higher

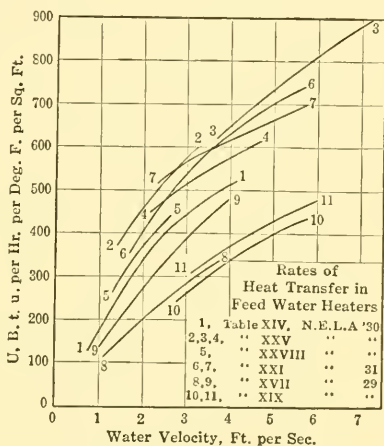


FIG. 950.—Heat Transfer Coefficients in Feedwater Heaters.

⁶ See "Draining and Venting Extraction Heaters," Power Plant Engineering, April 1, 1931, pp. 398-400.

⁷ See statement by J. I. Yellott, Report on Turbines by the Prime Movers Committee, N.E.L.A., July, 1932.

⁸ See "Closed Feedwater Heater Design," by Frank R. Wheeler, Power Plant Engineering, July, 1933, p. 289.

than those commonly used with plain tubes, heat transfer coefficients as high as 1000 B.t.u. per hr. per sq. ft. per deg. fahr. L.M.T.D.⁹ have been obtained in film-type heaters fitted with brass tubes. The great disadvantage of this form of heater is the necessity of using only feedwater of the highest quality in order that the annular spaces maybe kept clear.

Closed heaters, designed for use where space is restricted, are sometimes fitted with devices which are inserted within the tubes for the purpose of causing the water to follow a helical path through the tube. This tends to bring each particle of water more often into contact with the tube surface and thus to increase the rate of heat transfer.

(e) The **amount of heating surface** required in a closed heater depends upon (1) the quantity of heat to be transferred in any given time, (2) the overall coefficient of heat transmission for the apparatus, and (3) the mean temperature difference between the two fluids passing through the heater. The first item is fixed by operating requirements; the second must be evaluated from the physical and thermal characteristics of the equipment; the third is based upon these characteristics and upon economic considerations. A detailed explanation of this phase of heater design is given in Sects. 339 and 340, to which attention is directed.¹⁰

552. Adaptability of Open and Closed Heaters.—(a) In earlier power-plant practice the particular operating conditions for which each type of heater was especially fitted were considered to be well established, and, in general, the fields of usefulness did not overlap to any great extent. However, at the present time both open and closed heaters are in use for similar kinds of service, and the peculiar advantages of each are not so well defined as formerly.

(b) The open heater may be readily adapted to the processes of chemical purification and deaeration of feedwater; hence it is outstanding in this field. It is also particularly suitable for use in industrial plants where the condensate from process equipment, or the "returns" from a heating system, are piped direct to it and are heated by exhaust steam. An open heater must be set at such an elevation with respect to the inlet of its discharge pump as will insure the delivery of vapor-free liquid to the pump, the magnitude of this elevation depending upon the pressure and temperature of the water leaving the heater. In simple plants the discharge pump often functions also as the boiler feed pump, a second pump being required in some cases to force a supply

⁹ Logarithmic mean temperature difference.

¹⁰ See also "Closed Feedwater Heater Design," by Frank R. Wheeler, *Power Plant Engineering*, May, 1933, p. 200.

of feedwater to the heater. Occasionally an open heater of the jet type is arranged to combine the functions of a heater and of a condenser.

(c) Since the tube system of a **closed heater** is really a part of the feedwater piping, this apparatus may be placed on the high-pressure side of the boiler feed pump; and often only *one* pump need be operated in a feedwater system employing this type of heater. However, in a plant operating on the regenerative cycle this advantage may largely disappear, because additional pumps may be required both for handling the feedwater and also for transferring the condensate from the heaters to the feedwater circuit.¹¹ Closed heaters may be used with a wider variety of plant arrangements than open heaters since the steam and water need not be at the same pressure in the former type, while this requirement must always be considered with the latter. Usually a closed heater is smaller and more compact for a given capacity than an open heater, since the latter is, in most cases, designed for water purification as well as heating; however, if purification is not a factor in open-heater operation, this equipment may also be built in a very compact form.

(d) Feedwater heating on **steam locomotives** furnishes a good example of the adaptation of heaters to special operating conditions. Either open or closed heaters may be installed in connection with this non-condensing power plant, and each gives about the same results in service. In many recent designs either type is placed within the smoke-box of the locomotive, although standard practice has previously been to mount heaters externally. Each type is specially built to heat the feedwater from its temperature in the tender to an upper value ranging from 200 to 250 deg. fahr. The heating is done by means of a portion of the exhaust steam from the main cylinders in addition to all that discharged from certain auxiliaries, the total weight of steam used amounting to about $\frac{1}{6}$ of the weight of feedwater heated. In the open heater the condensate mixes directly with the incoming feedwater in the heater; when a closed heater is used, the condensate drains back to the tender. These heaters are built with capacities up to 80,000 lb. of water per hr., or more, and are capable of handling large quantities in a restricted space.

553. Feedwater Heating by Means of Special Equipment.—(a) In addition to the methods already described, feedwater may be heated through the use of various other kinds of **heat-recovery apparatus**, in some of which the temperature rise of the water is a secondary effect. Frequently the use of the feedwater as a **cooling medium** produces a considerable improvement in plant performance, and a number of

¹¹ See Power, Nov. 25, 1930, pp. 824-829 and 832-836.

arrangements involving this principle are employed. For example, in a large central station or industrial power plant the feedwater system may include such pieces of equipment as the generator air cooler, the bearing-oil cooler, the ejector condenser, the deaerator, and the evaporator condenser; and the feedwater is pumped through several or all of these devices in the proper sequence to obtain progressive heating.¹² When a system of continuous blowdown is used, the feedwater may receive heat from the blowdown water in a suitable heat exchanger, as described in Sect. 575 (d).

(b) The **steam injector**, which is used as a boiler feed pump in many small steam-generating plants and especially on steam locomotives, serves also as a very effective feedwater heater. It is capable of heating water to temperatures as high as 300 deg. fahr., depending upon its design and upon the temperature of the steam supplied to it; but its characteristics as a pump prevent its use in a system with other forms of feedwater heaters. Considered as a pump only, that is, as an engine for converting heat energy into work (see Sect. 560 (d)), the injector is a very inefficient device; but when viewed as a combined pump and feedwater heater its efficiency is almost 100 per cent, since all the energy of the steam jet is returned to the boiler except that lost to the atmosphere surrounding the injector and its piping. No thermal gain results in the plant as a whole when live steam is used as the driving and heating medium; however, *exhaust steam* from non-condensing engines may be used for this purpose, with a resulting increase in plant efficiency.

B. EVAPORATORS

554. Uses and Advantages of Evaporators.—(a) The **primary function** of an evaporator in a steam-generating plant is to vaporize raw or partly treated water, the vapor from which will subsequently be condensed for use as boiler feed or for process heating. A **secondary function**, and one which is of increasing importance, is that the apparatus may act also as one of the heat-recovery elements in a plant operating on the regenerative cycle. In some installations, notably marine power plants on ocean-going vessels, the supply of soft make-up water from natural sources is strictly limited, and in such cases evaporators are essential parts of the plant from the standpoint of safe operation. When the quantity of make-up required in a central station is below 5 per cent, and especially if it is less than 2.5 per cent, the installation of evaporators is usually justified economically, because the improvement in plant performance offsets the fixed and operating charges against this equip-

¹² See footnote 11 on page 947.

ment; in plants requiring over 5 per cent feedwater make-up it would probably be less expensive to treat all the make-up water completely by chemical processes, and to obtain good plant performance by means of some combination of apparatus which does not include evaporators.

(b) The following **advantages** are obtainable by the use of evaporators in steam-generating plants: (1) Since the feedwater is distilled, blowdown is used very infrequently, with a resultant reduction in the loss of both heat and water therefrom. (2) Scale formation on the boiler heating surfaces is eliminated. This results in a higher average heat-transfer rate within the boiler, and hence the full capacity of the units is available at all times, smaller units may be used, and the number of standby units may be reduced; also, operation at high pressures is safer, and there is a reduction in the amount of boiler cleaning required, as well as in the number of tube replacements. (3) Foaming is eliminated, with a resultant freedom from deposits of sediment in superheaters, prime movers, and piping. (4) The plant is independent of the public water supply since it can make use of cheap impure water from a river, a lake, or the ocean. (5) The soluble salts which remain in all chemically treated boiler water are eliminated; hence caustic embrittlement of the boiler plate is largely prevented. (6) Clean steam for process purposes can be produced efficiently, at a pressure lower than that maintained in the steam-generating units, without affecting the supply of boiler feedwater.

555. Classification of Evaporators.—The most fundamental basis on which evaporators may be classified is that relating to the disposition of the two fluids within the apparatus. On this basis the three recognized types are (1): the **steam-tube type**, in which the water to be vaporized surrounds tubes containing steam; (2) the **water-tube type**, in which the opposite arrangement exists; and (3) the **flash type**, in which preheated water vaporizes upon being injected into a chamber wherein the pressure is maintained at a value less than the saturation pressure at the temperature of the water (an example of this type is the **deaerator** which is discussed on page 985). Of the first two types, the steam-tube form is the one used almost exclusively in steam-generating plants; hence the water-tube type mentioned above will not be considered here. Steam-tube evaporators may be subdivided into (1) the **film type**, and (2) the **submerged-tube type**, each of which will be considered in detail later. An evaporator of the **pressure type** is one in which the steam is supplied to the apparatus at a pressure considerably above atmospheric, while one of the **vacuum type** receives steam at a pressure less than atmospheric.

An evaporating plant may be arranged to operate either in one stage or in several stages, the former arrangement being known as a **single-effect** and the latter as a **multiple-effect** plant. Multiple-effect plants may be **double-effect**, **triple-effect**, **quadruple-effect**, and so on. Although the number of stages could be increased indefinitely, there is a limit beyond which it is not feasible to go because of economic

considerations. This phase of evaporator plant design will be taken up in detail later.

556. Typical Arrangements of Evaporators.—(a) A **submerged-tube type** of evaporator is shown in Fig. 951; the through tubes used in this design are rolled into each of the tube sheets, and adequate provision for expansion and contraction is afforded by the slight initial

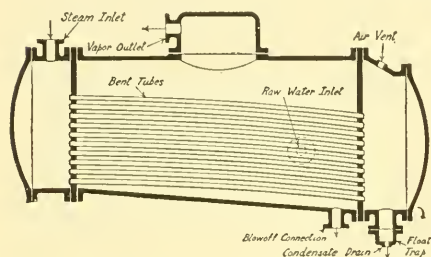


FIG. 951.—A Form of Evaporator Using Bent Through-Tubes.

curvature of the tubes, which allows them to “spring” when they are subjected to temperature changes. This change of shape also results in a natural tendency for the scale deposits to break away from the tube surfaces during normal operation. Since the tubes are in contact with water which may contain a high concentration of solids, as well as gases in solution, they should be made of corrosion-resistant material, commonly brass or copper.

(b) Another form of evaporator, which may be arranged for operation either as a **submerged-tube** or as a **film type**, is shown diagrammatically in Fig. 952. It is fitted with hairpin tubes which are rolled into each tube sheet and are supported at their outer ends in such a way that they are free to expand and contract. The shell is made of cast iron for low-pressure operation and of steel plate construction for high pressures. The collecting dome is fitted with baffles (not shown), the purpose of which is to prevent drops of water from becoming entrained with the vapor leaving the apparatus. The arrangement of this evaporator is such that any air which enters with the steam is pushed through the tubes into the condensate discharge space from which it is vented as

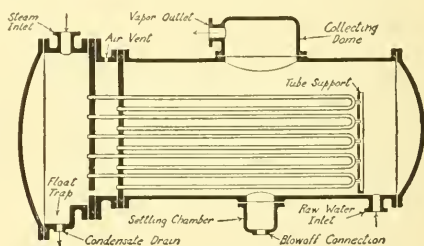


FIG. 952.—A Form of Evaporator Using Hairpin Tubes.

shown. The scale-shedding characteristics of the tubes are the same as those for the type of evaporator described in (a); any shattered scale finds its way into the settling chamber from which it is blown out to the sewer.

When this apparatus is operated on the submerged-tube principle, its water level is maintained at a point just above the top row of tubes, and make-up water is supplied at a rate equal to that of the formation of vapor. When the device functions as a film-type evaporator a pump well is included in the settling chamber; this provides for the injection of the raw water, and also makes possible the attachment of a suction pipe extending to a centrifugal pump which can draw water from the well and force it to a cascading device above each row of tubes. The water would then fall over the tubes in a thin film and any which failed to evaporate would return to the well to be re-pumped. A float-valve mechanism in the well is used to regulate the supply of raw water so that a definite level may be maintained in the well at all times. In this case the settling chamber is so baffled that scale cannot find its way into the pump suction. The chief advantage of the film type is that heat transfer from the steam to the water proceeds at a somewhat more rapid rate than it does in the submerged tube form; the chief disadvantage is the extra equipment and energy required to circulate the water.

(c) **Other tube forms and arrangements** are also used in power-plant evaporators. The object of other designs, as with those already considered, is to make adequate provision for the expansion and contraction of the tubes and, if possible, to make them self-cleaning on the water side. Thus evaporators are built with tube-coils made in the form of double spirals, these coils being placed one above the other in horizontal planes; another design is provided with a large number of helical tube-coils having vertical axes and extending from an upper steam header to a lower condensate header. The former may function as a film-type evaporator; the latter is arranged for submerged-tube operation.

557. Energy Equations for Evaporators.—(a) The transfer of heat from the steam to the water in an evaporator may be expressed in terms similar to those used in Eq. (649). On the assumption of steady flow and of no heat losses through the shell,

$$w_s(h_a - h_b) = w_w(h_2 - h_1), \quad . \quad . \quad . \quad . \quad (653)$$

where w_s and w_w are, respectively, the weights of steam and of water entering the evaporator, in lb. per hr.; h_a and h_b are, respectively, the

heat contents of the steam entering and of the condensate leaving, in B.t.u. per lb.; and h_2 and h_1 are, respectively, the heat contents of the vapor leaving and of the water entering, also in B.t.u. per lb.

(b) The **overall coefficient of heat transfer** (U) in an evaporator is influenced chiefly by the thicknesses of the fluid films and the presence of deposits on the heating surfaces. The latter is of great importance in evaporator practice, owing to the rapid formation of scale on the water side of the tube when impure water is supplied; however, little trouble is experienced from deposits on the steam side of evaporator tubes. The thickness of the liquid film is governed chiefly by the viscosity and velocity of the water; and the viscosity of water depends much more on temperature¹³ than is generally appreciated. In general, the coefficient of heat transfer increases as the temperature difference between the steam and the water increases. For any given temperature difference, the coefficient also increases with a rise in the temperature of vaporization. The numerical values of this coefficient for evaporators are slightly lower than those used in condenser and feedwater heater practice; however, the design of evaporators, and their arrangement within the plant, are subject to such wide variations that it is difficult to obtain representative values of the heat transfer coefficient for all conditions. A chart given by Badger¹⁴ shows that when the difference between the temperature corresponding to the pressure of the steam and that corresponding to the pressure in the vapor space was 40 deg. fahr., the coefficient of heat transfer had a value of 325 B.t.u. per hr. per sq. ft. per deg. fahr. L.M.T.D. when the temperature in the vapor space was 140 deg. fahr., and a value of 545 when this temperature was 212 deg. fahr. These figures were obtained from tests performed on an evaporator having *vertical tubes*, and are therefore useful only as an example of the results obtainable with this type of apparatus. Full consideration has been given to the subject of heat transfer in evaporators by several authors, to whose works attention is directed.¹⁵ Evaporators are designed to operate satisfactorily on temperature differences between the steam and the water ranging from 20 to 100 deg. fahr.

558. Evaporation in Single and Multiple Effects.—(a) When a power plant is being designed, the decision as to whether the evaporation of make-up should be done in single or in multiple effects rests upon

¹³ See Fig. 487, page 269.

¹⁴ See "Evaporators for Boiler-Feed Make-Up Water," Trans. A.S.M.E., FSP-50-45 (1928), pp. 207-212.

¹⁵ See "Heat Transfer and Evaporation," by W. L. Badger, the Chemical Catalog Co., Inc. (1926); "Evaporating, Condensing, and Cooling Apparatus," by Hausbrand and Heastie, D. Van Nostrand Co., Inc. (1929); and "Evaporation," by Webre and Robinson, the Chemical Catalog Co., Inc. (1926).

economic considerations associated with the energy balance of the plant. As stated above, so many variables enter into this problem that specific recommendations cannot be made in a text of this kind; if definite information is required, reference should be had to tests of evaporators¹⁶ and to typical diagrams of the cycles of working substances used in power plants.¹⁷ However, an analysis of the operation of single- and multiple-effect evaporator plants may serve to bring out the advantages and disadvantages of different arrangements, and may also enable general conclusions to be drawn.

(b) The following **numerical example** will cover single-, double-, and triple-effect evaporation, using an initial steam pressure of 25 lb. per sq. in. abs. and a condenser pressure of 3 lb. per sq. in. abs. in each case.

Assume that (1) the steam entering the first effect is dry and saturated, as is also the vapor produced in each effect; (2) raw feedwater is introduced into each effect; (3) the condensate from each effect enters the feedwater system at a particular point where the temperatures of the condensate and of the feedwater are equal; (4) a condition of steady flow exists; (5) there are no losses of heat through the shell or piping of any evaporator; and (6) in each case the results are obtained for 1 lb. of steam admitted to the first effect.

SINGLE EFFECT

The arrangement, operating conditions, and results obtained are shown in Fig. 953. From Eq. (653),

$$w_s(h_a - h_b) = w_w(h_2 - h_1),$$

or

$$1(1160.2 - 208.3) = w_w(1122 - 28).$$

Therefore,

$$w_w = 0.87 \text{ lb. of vapor produced per lb. of steam supplied.}$$

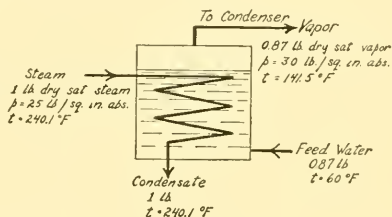


FIG. 953.

DOUBLE EFFECT

In this case an initial assumption must be made regarding the proportional temperature drops in the two effects. For the purposes of this example, assume that half of the total drop from the temperature of the steam entering the first effect to the temperature of the condenser occurs in each stage; actually the temperature drop in each effect tends to adjust itself automatically, so that the quantity of heat transferred per hour will be the same in each effect. This, in turn, depends upon the individual coefficients of heat transmission, the values of which are influenced by various complex factors. The result is that the temperature drops in the two effects will not be exactly equal; however, an analysis of this phase of evaporator design is beyond the scope of this work. Using the assumption just made, the

¹⁶ See the annual reports of the Prime Movers Committee in Trans. N.E.L.A.

¹⁷ Ibid.; see also Chapter XLIX, and Power, Nov. 25, 1930, pp. 825-829.

conditions and results are as shown in Fig. 954. The temperature in the vapor space of the first effect, and in the coils of the second effect, becomes 190.8 deg. fahr., and this is the saturation temperature for a pressure of 9.5 lb. per sq. in. abs., which then exists at these points. If w_1 and w_2 are, respectively, the weights of water vaporized in the first and second effects, per pound of steam admitted to the first effect, Eq. (653) may be applied again, as follows:

$$1(1160.2 - 208.3) = w_1(1142 - 28).$$

Therefore,

$$w_1 = 0.855 \text{ lb.}$$

The steam admitted to the coils of the second effect is the 0.855 lb. produced in the first effect. Then,

$$0.855(1142 - 158.7) = w_2(1122 - 28).$$

Therefore,

$$w_2 = 0.768 \text{ lb.}$$

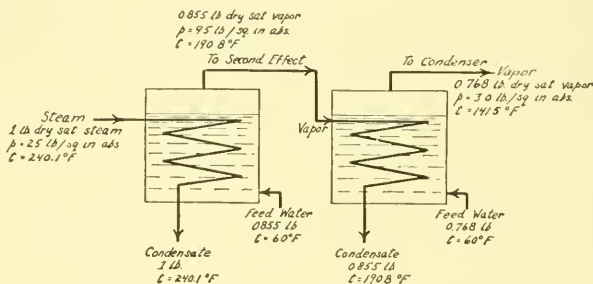


FIG. 954.

The total quantity of condensate resulting from the vapor produced in both effects, per pound of steam admitted to the first effect, is

$$w_w = w_1 + w_2 = 0.855 + 0.768 = 1.623 \text{ lb.}$$

TRIPLE EFFECT

In this case the assumption is made that one-third of the total temperature drop occurs in each effect, the conditions and the results obtained being shown in Fig. 955. The temperature in the vapor space of the first effect becomes 207.2 deg. fahr., and that in the vapor space of the second effect becomes 174.3 deg. fahr., with corresponding pressures of 13.4 and 6.7 lb. per sq. in. abs. at these points. Applying Eq. (653) as before, and using w_3 to represent the weight of water vaporized in the third effect, per pound of steam admitted to the first effect,

$$1(1160.2 - 208.3) = w_1(1148.5 - 28).$$

Therefore,

$$w_1 = 0.85 \text{ lb.}$$

The vapor used in the coils of the second effect is the 0.85 lb. produced in the first effect. Then,

$$0.85(1148.5 - 175.2) = w_2(1135.7 - 28).$$

Therefore,

$$w_2 = 0.747 \text{ lb.}$$

The vapor used in the coils of the third effect is the 0.747 lb. produced in the second effect. Then,

$$0.747(1135.7 - 143.1) = w_3(1122 - 28).$$

Therefore,

$$w_3 = 0.678 \text{ lb.}$$

The total weight of condensate resulting from the vapor produced in all three effects, per pound of steam admitted to the first effect, is

$$w_w = w_1 + w_2 + w_3 = 0.85 + 0.747 + 0.678 = 2.275 \text{ lb.}$$

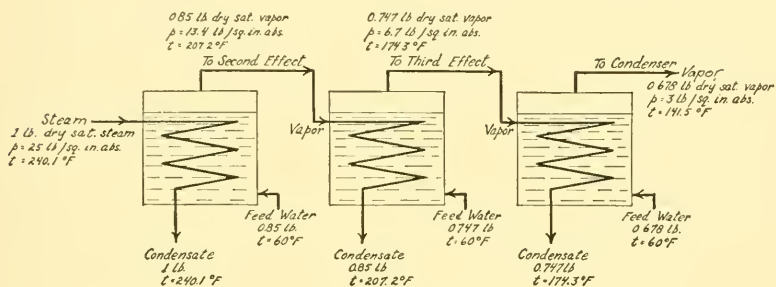


FIG. 955.

(c) Several important points are brought out by a study of this example, viz.: (1) As more effects are added, the **economy** of the evaporator plant increases; that is, more vapor is produced for each pound of steam admitted to the first effect. (2) Under the assumptions made for this example, the coefficient of heat transfer and the area of the heating surface for the single-effect evaporator are respectively equal to the coefficient and the area for *each* of the evaporators of the double- and the triple-effect. This being the case, the quantity of heat transmitted per hour per square foot of heating surface in *each* evaporator is directly proportional to the mean temperature difference between the steam and the water (see Eq. (432), Sect. 339). In the double-effect this temperature difference is half, and in the triple-effect it is one-third, of that in the single-effect; hence each unit of area in the double-effect transmits only one-half, and in the triple-effect only one-third, as much heat per hour as does each unit of area in the single-effect. Therefore the **capacity** of *each* evaporator (that is, the quantity of heat transmitted per hour per square foot) varies inversely as the

number of effects, the rate of transfer in the single-effect being taken as the standard. (3) The double-effect produces 0.753 lb. of vapor *more* than the single-effect, per pound of steam admitted, and the triple-effect produces 0.652 lb. *more* than the double-effect. The increase in vapor production therefore proceeds at a declining rate as more effects are added; hence an **economic limit** will be reached beyond which the thermal gain due to the increased number of effects is outweighed by the fixed and operating charges on the additional equipment. (4) The statement is sometimes made that, in a multiple-effect plant, 1 lb. of steam admitted to the first effect will result in the evaporation of as many pounds of water as there are effects. This principle was evolved at a time when almost no information existed regarding the properties of water and steam; obviously it is not well founded, and its use does not result in even a rough approximation in most cases.

(d) Various **methods of feed** are used in multiple-effect plants for the general purpose of obtaining a thermal gain; but such methods are of particular interest only in certain industries, such as sugar refineries, where special solutions are being handled. Thus, there is *parallel feed*, which is similar to that used in the example above; *forward feed*, in which the raw liquid is fed to the first effect and, after partial evaporation, flows to the next effect, and so on in series, the concentration of the solution increasing steadily; and *backward feed*, which is the opposite of forward feed. These methods have been treated extensively in various works on evaporation.¹⁸

559. Adaptability of Evaporators.—(a) From a thermal standpoint an evaporator gives very good performance because all the energy supplied to it with the steam, except that lost to the surrounding atmosphere and that lost in the blowdown, appears in the vapor produced, or in the condensate. Therefore both the vapor and the condensate should be handled in such a way that their energies will be conserved to the greatest possible extent. The vapor should be condensed by the feedwater which is on its way to the boiler, so that the latent heat of vaporization of the former may be returned to the boiler; and the condensate should enter the feedwater system at a suitable point depending upon its temperature.

The evaporator itself can also be arranged to give the equivalent of regenerative feedwater heating. If steam is bled from an intermediate stage of an engine or turbine and used in an evaporator, and if the vapor thus formed is condensed in the equivalent of an extraction heater, the result is almost the same as though the bled steam had been used directly in the heater. The result would be actually the same except

¹⁸ See footnote 15, p. 952.

that the terminal temperature difference (that is, the difference in temperature between the steam leaving the prime mover and the water leaving the heater), will be greater when the evaporator is interposed. The method is very convenient, and possesses the following advantages: (1) in an emergency, the capacity of the evaporator can be increased by connecting its vapor outlet to a heater having a lower pressure in the steam space than the one generally used; (2) the gas driven off from the make-up water in the evaporator finds its way down through the feed heating equipment to the main condenser, from which it is extracted by means of the air pump. Common practice in many plants is to heat the make-up water before admitting it to the evaporator, in order to drive off any gases which, if present, might result in the corrosion of parts of the evaporator itself.

Evaporators may be fitted into the power plant in a great many different ways, and should be installed so as to give the most economical, or the most convenient, result. In general, the required heating surface will decrease as the temperature difference between the steam and the water increases; therefore, other factors being equal, the evaporator would cost the least when the steam is supplied to it at the highest available pressure and the vapor is produced under the lowest possible pressure.

(b) Evaporators may be used very effectively as a means of producing **process steam** at a pressure lower than that maintained in the steam-generating units. In such installations the steam supplied to the evaporator coils is generated at a high pressure, and is first expanded through a turbine, from which it flows into the coils at a temperature sufficient to cause the production of process steam, in the evaporator, at the required lower pressure. The condensate from the evaporator then forms part of the boiler feedwater. One such installation,¹⁹ forming part of a central station, consists of seven evaporators, each receiving its proportionate share of the exhaust steam from a 12,500-kw. turbine. At rated load, about 530,000 lb. of exhaust steam reach the evaporator coils per hour at a maximum pressure of 400 lb. per sq. in. and give up sufficient heat to produce 400,000 lb. of process steam per hour at a pressure of 180 lb. per sq. in. In this case the process steam is subsequently raised to a temperature of 440 deg. fahr. in a live-steam superheater; and the condensate from the process steam is returned to the evaporators at a temperature of about 180 deg. fahr. The process plant circuit is thus a closed one, and is entirely separate from the main plant circuit. The chief advantage of this arrangement is that the contamination of the feedwater in the main plant, in which steam is

¹⁹ See Power Plant Engineering, Nov. 15, 1929, pp. 1204-1214.

generated at a pressure of 1200 lb. per sq. in., is reduced to a minimum; thus the 1200-lb. system approaches the condition of a perfectly closed cycle. In a plant of this kind, which produces large quantities of process steam and requires considerable make-up, it is essential that the evaporators be maintained in the same way as the main boilers; hence all the make-up should be carefully treated in order that scaling of the evaporator tubes may be avoided.

(c) When raw water is distilled in an evaporator plant, the concentration of solids in the liquid which is being evaporated increases steadily. Some system of **blowdown** must therefore be used, in order that this concentration may be controlled. The blowdown system may be either intermittent or continuous, the general arrangement and operation being essentially the same as that used in connection with boilers (see Sect. 575, p. 979). In a number of plants the water fed to the evaporators undergoes a preliminary chemical treatment in order that some of the suspended and dissolved solids may be precipitated outside of the evaporator and that excessive formation of scale within the shell may be prevented. In many cases, also, the feedwater for the evaporators is preheated in order that the performance of both the evaporators and the entire plant may be improved.

C. PUMPS, TRAPS, AND SEPARATORS

560. Types of Pumps.—(a) The various types of pumps that may find application in steam plants may be classified as follows: (1) **positive-displacement reciprocating pumps**, in which a piston, plunger, or bucket acts within a pump cylinder; (2) **positive-displacement rotary pumps**, in which the fluid is impelled by rotating pistons, lobes, gears, or screws; (3) **centrifugal pumps**, having a rotating impeller and not necessarily delivering an amount of fluid in fixed proportion to the impeller displacement; (4) **jet pumps**, in which the fluid pumped is propelled by a jet of the same or other fluid, as in the case of the steam injector; and (5) **direct-pressure pumps**, in which the surface of the fluid pumped is acted upon directly by another fluid under pressure, as in “pulsometers” and “air lifts.” Only the reciprocating pumps, centrifugal pumps, and injectors will be given further consideration.

(b) **Reciprocating pumps** can be subdivided as follows: (a) **direct-acting pumps**, which are without cranks, flywheels, and connecting rods, as exemplified by the direct-coupled condensing-water pump and its steam cylinder shown in Fig. 904; (b) **flywheel pumps**, which have flywheels, cranks, and connecting rods to control definitely the motion of their pistons; (c) **power pumps**, the crankshafts of which are driven

by means of belts and pulleys or by gearing interposed between them and the source of motive power. A reciprocating pump with a single pump cylinder is designated as a **simplex pump**; one with two cylinders as a **duplex pump**; and one with three cylinders as a **triplex pump**.

Pistons in pumps of this general type may be fitted with soft packing (e.g., "hydraulic," or cup-leather) if cold water is handled, but metallic piston rings or merely water grooves are often used when hot water is pumped. Plungers may be *inside-packed* or *outside-packed*, the accessibility of the packing in the latter case being particularly advantageous in high-pressure pumping. **Single-acting pumps** have plungers which enter the cylinders through outside-packed stuffing boxes; **double-acting pumps** may have either pistons, as in Fig. 904, or single or double plungers. When plungers are used, the cylinders have transverse internal partitions at mid-length to divide them into two independent parts, each end being equivalent to the cylinder of a single-acting pump. An **inside-packed pump** has a single plunger which reciprocates within a long sleeve and a stuffing box formed in the partition, the stuffing box lacking ready accessibility. The so-called **outside-packed pump** has two single-acting opposed plungers entering the opposite ends of the divided cylinder, the two plungers being connected by means of external cross yokes and rods. In the latter case the transverse partition in the cylinder is solid and no cylinder heads are needed. These outside-packed pumps are very suitable for high-pressure service.

(c) **Centrifugal pumps** are either of the **volute type** with scroll-shaped casings, as in the condensate and circulating pumps already considered, or of the **turbine type**, in which there is an annular *diffusion ring* with guide or diffusion vanes for properly directing the water after it leaves the rim of the impeller. Centrifugal pumps may be either single-stage or multi-stage; *single-stage pumps* usually have only one impeller, with either a single or a double suction; *multi-stage pumps* consist of two or more impellers handling the fluid in series to give the high pressure desired.²⁰ For large capacities two or more impellers, arranged to handle the fluid in parallel, may be mounted on the same shaft and enclosed in the same casing or in separate casings.

(d) **Steam injectors**, some aspects of which were discussed on page 948, act on the same principle as the steam-jet air ejector, but handle boiler feedwater and have only one injection nozzle in each stage. In the most elementary form of an injector (see Fig. 956) steam passes through the *nozzle* and *combining tube* and discharges through an overflow to the atmosphere until the device becomes primed; as soon as the

²⁰ See "Centrifugal Boiler Feed Pumps," by J. G. Mingle, Steam-Plant Engineering, February, 1932.

feedwater arrives in the combining tube it is projected across the overflow gap and passes through a diverging *delivery tube* and a check valve to the boiler, the only moving parts being the valves. The function of the combining tube is to increase the velocity pressure of the moving mass of fluid, while that of the delivery tube is to convert this high velocity pressure into static pressure of sufficient magnitude to lift the check valve and enable the water to flow into the boiler against the pressure existing therein.²¹

Injectors may be classed as **lifting** or **non-lifting**, depending upon whether or not they are capable of raising their water supply from a level lower than that of the combining tube; and either of these types is used in steam locomotive service as the main piece of equipment for feeding the boiler whenever the resistances to fluid flow in the high-pressure boiler feed line are not great enough to necessitate the use of reciprocating or centrifugal pumps.²² **Automatic injectors** are fitted with devices which cause the apparatus to restart itself after a temporary interruption of the water supply to the combining tube; and

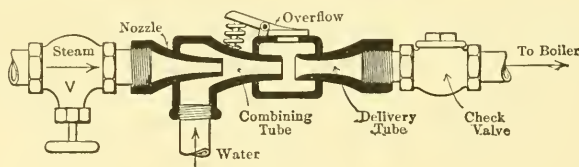


FIG. 956.

hence such injectors are particularly suitable for use in small stationary plants and on traction, logging, and road engines where the operator may have so many duties that he cannot give unfailing attention to the feeding of the boiler.

In the **double-tube positive type** the water being pumped is handled in two stages arranged in series, each with its own nozzle and delivery tube; and in such designs the first stage is supplied with steam at a pressure only slightly above atmospheric, usually a portion of the exhaust steam from a non-condensing engine, while the second stage uses live steam. Although the exhaust steam used may be at a pressure of about one atmosphere, the static pressure of the fluid leaving the first-stage delivery tube may be five or six atmospheres, while the static pressure of

²¹ For a further explanation, see "Practice and Theory of the Injector," by S. L. Kneass, John Wiley & Sons, 1910; also see "Characteristics of Injectors," by R. M. Ostermann, Trans. A.S.M.E., RR-51-3, 1929; for operating data see "Operation of an Automatic Injector," by V. R. Gage, Power, Feb. 27, 1917, p. 281.

²² See the "Locomotive Cyclopedia," Simmons-Boardman Publishing Co.

the fluid leaving the second stage must be higher than the pressure in the boiler being fed. This type of injector functions properly over a wide range of operating conditions, and is able to handle somewhat hotter water against a higher delivery pressure than is possible with the simpler arrangement. Its operation requires the use of several valves which must be manipulated in the proper sequence, and these must be carefully adjusted to maintain the desired rate of flow.

561. Boiler Feed Pumps.—(a) The **type of pump** to be used in forcing the feedwater into the boiler depends upon the capacity and the physical characteristics of the plant, and upon the operating conditions involved. Formerly, direct-acting simplex or duplex pumps, triplex power pumps, and injectors were the only kinds used, and these types are still commonly employed in small and moderate-sized installations; however, in the large modern plants multistage centrifugal pumps are

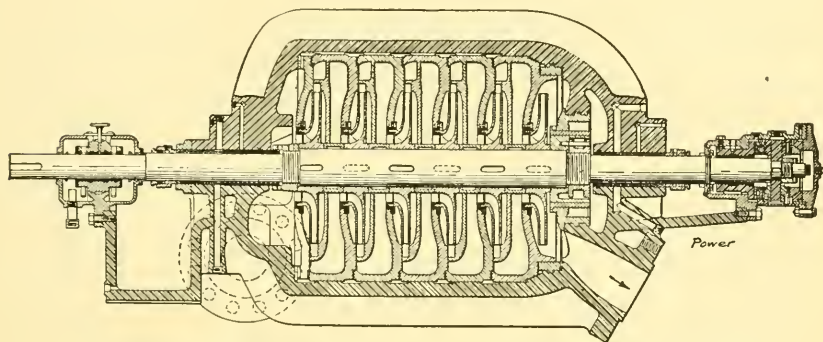


FIG. 957.—A Six-Stage, High-Pressure Centrifugal Boiler Feed Pump.

now used almost exclusively, although their employment in high-pressure service (e.g., 1200 lb. per sq. in.) has occasionally been attended by operating difficulties, with the result that reciprocating pumps are being substituted for them in some cases. As compared with centrifugal pumps, reciprocating pumps operate at higher efficiencies in boiler feed service although, for a given capacity, they cost more, occupy a greater floor space, and require more maintenance; furthermore, in high-pressure service they are likely to produce pulsations in the feed piping because the air chambers, which are fitted for the purpose of eliminating the pressure variations in the line, are not entirely effective under these conditions.

(b) **Centrifugal boiler feed pumps**, one form of which, for pumping against 1600 lb. per sq. in., is shown in Fig. 957, may be driven by constant-speed or variable-speed motors, or by steam turbines; in some

cases a pump is fitted with both a motor and a turbine, this arrangement being termed a "dual drive." These pumps usually have from two to seven stages, depending upon the magnitude of the discharge pressure, although single-stage pumps (e.g., the Bethlehem-Weir type) are made for discharge pressures as high as 800 lb. per sq. in. Satisfactory centrifugal pumps for high-pressure service can be designed only by accepting a compromise between the most desirable mechanical and structural features and the correct application of the principles of hydraulics;²³ and the selection of the proper type of pump for a given kind of service is dependent upon an analysis of the performance curves obtained from test data.²⁴

(c) The **control of the rate of boiler feeding** is usually accomplished by means of valves, operated either manually or by means of a feedwater regulator, at the boiler. In order that this control may be satisfactory, the pressure in the main feed line must be held constant, or it must be maintained at a constant differential above the steam pressure in the boiler, these functions being performed by a system of **governing** which is used to regulate the rate of discharge from the pumps. If a constant differential in pressure is to be maintained by means of centrifugal pumps driven at a constant speed, their discharge lines may be fitted with reducing valves, actuated by the boiler pressure, for controlling this differential; or if the discharge pressure is to be maintained constant at all times, unloading valves may be used in the discharge lines to by-pass sufficient water from the discharge to the suction of the pumps to produce the desired result. The rate of discharge may also be varied by changing the pump speed; if the pumps are motor-driven, speed can be regulated by the use of slip-ring induction motors or of direct-current motors with rheostat control; if the pumps are fitted with steam-turbine drives, the speed of the units can be controlled by means of governors of the "follow-up" type which adjust the steam supply to the turbines to counteract pressure changes in the pump discharge lines. The pressure at the pump discharges must be greater than that within the boilers by the amount required to overcome gravity and the resistances in the lines; and this means that the pressure at the pumps is likely to be 10 per cent, or more, higher than the pressure in the boilers.

²³ See "Meeting Hydraulic and Mechanical Requirements in Boiler Feed Pump Design," by O. H. Dorer, *Power*, May 28, 1929, p. 876.

²⁴ See "The Performance Characteristics of Centrifugal Boiler Feed Pumps," by J. G. Mingle, *Combustion*, July, 1932, p. 15, and August, 1932, p. 13; see also "Factors Involved in the Selection of Pumps," *Power Plant Engineering*, Oct. 1, 1930, p. 1124.

(d) The location and number of the boiler feed pumps depend upon many factors of plant design; hence this subject cannot be considered in detail here. Owing to the essential nature of the service rendered by these pumps, it is necessary to install reserve pumps for emergency operation; and several of the pumps should be steam-driven unless electric energy is also available from a reliable outside source. In many installations, some of the pumps are fitted with both motor and turbine drive. In every plant there should be more than one delivery line between the pumps and the boilers, and more than one source of feedwater for the pumps.

In order that plant operation may be satisfactory when the boiler feed pumps handle hot water, these pumps must receive their supply under a considerable pressure; and in most cases high-pressure pumps (e.g., those operating at 1400 lb. per sq. in.) receive their supply from "booster" pumps. Even a moderate feedwater temperature has a considerable influence upon the allowable "suction lift" of the pumps; centrifugal pumps are especially sensitive to changes in the temperature of their water supply, and care must thus be taken to locate them so that they will have the most favorable suction lift, otherwise their efficiency and capacity may be materially reduced. When hot water (say above 150 deg. fahr.) is pumped it should be supplied under a positive head, the magnitude of which depends upon the temperature and altitude.²⁵

562. Drainage of Feedwater Heaters and Piping.—(a) The drainage of condensation from pipe lines, heaters, and other apparatus may be accomplished in various ways, depending upon the physical characteristics of the plant; these ways are (1) directly, by gravity; (2) by free discharge from a high-pressure vessel to a low-pressure one or to the atmosphere, perhaps through a U-bend; (3) by using pumps; or (4) by means of traps or equivalent devices. The first method needs no discussion. Discharge through a U-bend to a vessel having a lower internal pressure has been illustrated by the drainage of condensers used with steam-jet air ejectors; in some cases the condensate from heaters used in the regenerative cycle is "flashed" directly into other heaters or vessels having a lower internal pressure, traps or U-bends often being included in this arrangement. Sometimes condensate from regenerative heaters is pumped to heaters or other vessels having a higher internal pressure, or is handled through a booster pump to the suction of the boiler feed pump. The traps, separators, and drips used

²⁵ See "Suction Lifts for Pumps," by D. McFee, *Power*, June 14, 1932, p. 876; also "Efficient Suction Head for Centrifugal Pump Operation," by A. Peterson, *Power*, August, 1932, p. 69.

in connection with the various drainage methods, though apparently insignificant, are important pieces of equipment, the failure of which may lead to plant shutdowns, or other serious difficulties, with resulting expense.

(b) **Steam traps** are automatic devices used to remove condensate from power-plant equipment without allowing the escape of steam from the equipment. In general, traps are either of the **return type**, which includes all those designed to return the condensate directly to the boiler, or the **non-return type**, which are designed to discharge to the atmosphere or to receptacles having a lower internal pressure than that of the boiler. Traps may also be classified as (1) **high-pressure**, (2) **low-pressure**, and (3) **vacuum**; and all of these may be further classified as follows, according to the principle on which they operate: (a) **float traps**, into which the water gravitates and lifts a float, thus opening a valve and allowing the water to discharge under the pressure within the float chamber; (b) **bucket traps**, in which the water rises until it overflows the edges of a bucket which then sinks, thereby opening a valve and allowing the water to discharge from the bucket under the pressure within the trap; (c) **dump or bowl traps** having counterweighted closed bowls mounted on one end of a short length of pipe which is pivoted at the other end on hollow trunnions containing valves; when the closed bowl receives enough water to overcome the counterbalance, it descends, shutting the drainage valve, opening the discharge valve, and admitting live steam to displace the water; (d) **expansion traps**, which rely upon the relative expansion of two metals or on the action of a volatile liquid behind a flexible diaphragm to open the discharge valve when the temperature of the collected water becomes sufficiently reduced below that of the steam; (e) **differential traps** having the discharge valve controlled by a diaphragm, one side of which is exposed to the steam pressure and also to a sealing water column of fixed height, and the other side to the column of collected water—when the amount collected becomes sufficient to overcome the pressure of the steam and the sealing column, the discharge valve opens; (f) **siphon traps**, employing an inverted U, as already discussed; and (g) the **steam loop**. This loop has a well-insulated riser extending vertically upward from the bottom of a collecting pocket in the main steam line and discharging from above into one end of a long bare horizontal pipe which has its other end connected to the lower part of the boiler drum by means of a descending pipe known as a drop leg; by draining the drop leg, steam is allowed to flow through the entire device from the main line, and some of this steam condenses in the bare horizontal pipe, reducing the pressure in this pipe and thus causing a mixture of spray and steam

to flow up the riser from the collecting pocket; the condensate from the horizontal pipe collects in the drop leg from which it is free to descend into the boiler drum as soon as its level rises high enough above that of the water in the drum to produce a hydrostatic pressure equal to the small difference in steam pressure between the boiler and the collecting pocket in the main line; after the initial draining of the drop leg, the action becomes continuous and the device requires no further attention; one arrangement of this device is known as the *Holly loop*.²⁶

(c) **Steam separators** are pockets placed in steam lines to remove from the flowing steam most of the liquid entrained therein. They are used in main pipe lines when the steam is saturated or may become so, and in exhaust-steam lines when the steam contains oil which would be detrimental if present in the condensate. These devices may be built to operate on any of the following principles: (1) a **reverse-current**, or abrupt 180-deg. change in direction may be used to throw the liquid out of the current of steam; (2) **centrifugal action**, caused by a rotary motion imparted to the steam and entrained liquid, may be used in a similar manner; (3) **baffle plates**, usually corrugated or fluted, may be so placed in the path of the steam that the particles of liquid will adhere to them and be drained off by gravity; or (4) **mesh** may be used to effect mechanical filtration. The liquid which is separated from the steam is usually collected in drip pockets from which it is discharged by means of steam traps or other similar automatic equipment.

²⁶ For further data on traps, see "Steam Traps—Their Selection, Installation, and Upkeep," by E. Smiley, *Power*, July 11, 1922, p. 45; "Steam-Trap Installation and Operation," by R. N. Robertson, *Power*, Oct. 9, 1923, p. 573; also "New Steam Traps for High Pressures and Large Discharge," by C. H. S. Tupholme, *Combustion*, Oct. 1931, p. 22.

CHAPTER XLVI

BOILER WATER CONDITIONING ¹

563.—General Characteristics of Natural Waters and the Determination of Impurities.—(a) Many impurities, in the form of *gases*, *liquids*, and *solids*, dissolve readily in all natural waters; consequently the water supplies taken from streams, lakes, and wells always contain substances in solution, in addition to whatever finely divided organic and mineral matter may be carried in suspension. Before these “raw” waters can be successfully used for the generation of steam in industrial and power-plant boilers they must usually be conditioned to eliminate any impurities which would cause dangerous or other undesirable operating conditions. The important impurities present in water intended for use in boilers must be determined before the proper conditioning can be effected.

(b) **Definitions, test methods** used in determining impurities, and the **characteristics** imparted to water by these impurities are given in the discussion immediately following:

(1) The **total solids** present in the water are defined as the material remaining when a sample of water is evaporated to dryness; and the quantity is expressed either in parts per million (p.p.m.), or in grains per U. S. gallon. By “parts per million” is meant the weight of the total solids (expressed in any unit, such as pounds or grams) per million units of weight of pure water. Grains per U. S. gallon is self-explanatory. (Parts per million may be converted into grains per U. S. gallon by multiplying the former by 0.0583. In some cases water chemists report results in parts per 100,000, or in pounds per 1000 U. S. gallons.)

(2) **Suspended solids** is the term used to denote the solid material carried by water, i.e., material which is not in chemical solution. The quantity of suspended solids present may be determined by filtering a sample of the water through a filter paper of known weight and comparing the weight of material retained on the paper with the weight of the sample filtered. An alternative method is to evaporate to dryness equal quantities of filtered and unfiltered water, the difference in weight of the residues being taken as the total suspended solids. This quantity is expressed in the terms stated in (1).

(3) The **dissolved solids** are defined as the residue obtained after the evaporation to dryness of a filtered sample; and the quantity is expressed in the terms given in (1). (Note that the weight obtained by evaporation to dryness may or may not indicate the *total* material dissolved in the water, since any gas in solution will necessarily be driven off, as will almost all other liquids that may be in solution; also, certain dis-

¹ The conditioning of condensing water was discussed in Sect. 544, p. 930.

solved solids may decompose during the evaporation and subsequent drying of the residue, and liberate gases which pass off.)

(4) **Hardness of water** is the expression employed when soap is used unsuccessfully in washing with such water. Waters containing relatively large quantities of calcium and magnesium and certain other materials constitute hard waters. The ingredients causing the hardness react with soap, and no lather or suds can be formed until this action is practically complete. Water is said to be softened when these materials have been more or less completely removed by chemical treatment. The **total hardness** of water is measured by the amount of a standard soap solution used up in reacting with the constituents causing hardness; it is expressed in the terms stated in (1), and is calculated backward from the known characteristics of the soap solution, the result being only approximate at best.

(5) **Temporary, or carbonate, hardness** is the term used to indicate the presence of the bicarbonates of calcium, magnesium, and some other materials. These salts are said to create temporary hardness because they are converted very largely to monocarbonates, or "normal carbonates," by boiling, and then become only very slightly soluble; that is, the salts causing temporary hardness can be almost completely precipitated by boiling the solution, the water then becoming relatively soft. Carbonate hardness is expressed in the terms stated in (1).

(6) **Permanent, or non-carbonate, hardness** is the hardness due to the presence of calcium and magnesium salts, and a few others, of such character that they cannot be precipitated out of solution by simple boiling. These salts are generally the sulphates. Non-carbonate hardness is expressed in the terms stated in (1).

(7) **Total alkalinity** is the term used to designate the weight of all forms of alkaline salts in a sample of water. These are principally the carbonates, bicarbonates, and hydrates. The total alkalinity is determined by measuring the quantity of an acid, such as H_2SO_4 , required to neutralize the sample. Total alkalinity is generally expressed as the amount of calcium carbonate, or of sodium carbonate, equivalent to the H_2SO_4 used, and in the units stated in (1). It should be noted that when the feedwater chemist speaks of calcium carbonate or sodium carbonate without modification, he means the normal or monocarbonate. In some cases the type or types of salt responsible for alkalinity must be determined. When this is done, reference is made to carbonate alkalinity, bicarbonate alkalinity, and hydrate alkalinity. The units of measurement are the same as for total alkalinity.

(8) The **total acidity** is indicated by the total quantity of material present which can be neutralized by a base, such as sodium hydroxide. It is expressed indirectly in parts per million, or other equivalent, of calcium carbonate that it is theoretically capable of neutralizing. For some purposes it is desirable to determine the acidity due to different materials (such as the free mineral acids); in all such cases the determination is made by finding the quantity of some base, such as sodium hydroxide, required to neutralize the sample and then to express this quantity indirectly in terms of calcium carbonate, as stated above.

(9) The **hydrogen-ion concentration, or pH value**, is a method of expressing either the acidity or the alkalinity of liquids, and is based on simple tests which are much more sensitive and give more exact values than do the neutralizing methods described above, since the latter give only approximate results with most of the solutions encountered in practice.

A brief explanation of the **principle of hydrogen-ion concentration** is as follows: There is a certain amount of dissociation in all water solutions, and the degree of balance between the H-ions and the OH-ions determines the character of the solution.

The H-ions are positively charged and give an acid reaction; the OH-ions are negatively charged and give a basic reaction; consequently if the former predominate, the solution is acid, and, if the latter, the solution is basic. Determination of the hydrogen-ion concentration, or *pH* value, therefore gives a ready means of discovering whether a sample of water is acid or basic.

The *pH* value is stated in terms of numbers based on a logarithmic scale, in accordance with which a condition of neutrality of the water at 72 deg. fahr. is given by the number 7. This figure is used because one liter of pure water at 72 deg. fahr. contains $1/10^7$ gram of ionizable hydrogen, the number 7 thus being the common logarithm of the *reciprocal* of the fractional quantity just given. Therefore the *pH* value of pure water, at any temperature, is really the common logarithm of the *reciprocal* of the H-ion concentration at that temperature. The *pH* value corresponding to neutrality undergoes marked changes with temperature, as shown in Table LXXIX. Acid solutions give a lower *pH* value than that which corresponds to a neutral condition at the temperature of the water, and alkaline solutions a higher value.

TABLE LXXIX

VARIATION OF NEUTRAL POINT WITH TEMPERATURE

Temperature, Deg. Fahr.	<i>pH</i> Value Corresponding to Neutrality
72	7
100	6.75
200	6.20
300	5.87
400	5.70
480	5.65 (minimum)
500	5.70
600	5.85

The measurement of *pH* values can be made by either the potentiometric or the colorimetric method; the former is essentially a laboratory method owing to its extreme sensitivity, and the latter is therefore to be preferred for making the determinations involved in routine control. Measurements by means of the colorimetric method depend upon the fact that given dyes, called indicators, will impart definite shades or depths of color to solutions having certain *pH* values, each dye giving a gradation of shades over a small range of *pH* values. Routine testing consists in adding various dyes to the water solutions and comparing the resulting colors with those found in solutions of known *pH* value which were made with the corresponding dyes.²

(10) The **chloride content** of water may consist principally of sodium chloride, which is a very stable salt, and magnesium chloride, which is easily dissociated. The quantity of chloride is determined chemically, generally with silver nitrate solution, and is expressed in parts per million, or the equivalent.

(11) The **sulphate content** of raw waters is of importance for several reasons.

² See "Control of *pH* in Steam-Plant Practice," by F. R. McCrumb, Steam-Plant Engineering, September, 1932, p. 19.

In combination with calcium it forms calcium sulphate, the principal constituent of hard boiler scale; also, under certain conditions, definite ratios of sulphate with respect to other constituents must be maintained in the boiler water, as will be shown later for one case. The sulphate content of raw water, and of the water in the boiler, is thus one of the controlling factors in determining and regulating feed-water treatment. Determination is made chemically and the sulphate content is expressed either in parts per million, or the equivalent, or else as a ratio with respect to carbonates, as will be explained later.

(12) The **total gas content** of water is generally assumed to mean the quantity of gas which is dissolved, plus that which is mechanically entrained. However, the term is indefinite, since the water may contain salts, such as the bicarbonates, capable of liberating gases when the temperature is increased.

The gas which is mechanically entrained will separate out almost entirely if the sample is allowed to stand without agitation at a constant temperature. The gas which is dissolved can be liberated to any desired degree by raising the temperature of the water or by decreasing the concentration of the space above the water with respect to that gas. Dissolved gas can also be determined by means of chemical reactions which take place at the expense of that particular gas.

The quantity of either mechanically entrained gas or dissolved gas present in a sample of water is expressed in cubic centimeters per liter of water, or in parts per million; and when the volumetric unit is employed, the results are usually reduced to a pressure of one standard atmosphere and a temperature of 32 deg. fahr. On this basis a reading given in parts per million may be converted to cubic centimeters per liter by multiplying the given reading by a factor, as follows: For oxygen, 0.7; for carbon dioxide, 0.509; and for hydrogen, 11.24.

(13) **Dissolved oxygen**, even though present in very small quantities, makes water violently corrosive in certain power-plant apparatus under some conditions. The oxygen content is generally determined chemically and expressed in cubic centimeters per liter of water, or in parts per million.

(14) **Carbon dioxide** gas may exist in water in three different forms: (1) as free carbon dioxide in solution and possibly also mechanically entrained, (2) as part of a bicarbonate salt, or (3) as part of a carbonate salt. Since the bicarbonate readily gives up half of its carbon dioxide and forms the carbonate, half of the carbon dioxide present in the bicarbonate is called "half-bound" carbon dioxide; thus the carbon dioxide present as carbonate and half of that present as bicarbonate are held in relatively strong combinations. The sum is known as the "bound" carbon dioxide.

The amount of carbon dioxide present in water in the different forms is frequently of great importance in steam-engineering practice, both because of its effect upon feedwater behavior and treatment, and because of its corrosive effects under certain conditions. It is determined chemically and expressed in parts per million, or other equivalent measure.

(15) The **electrical conductivity** of chemically pure water is exceedingly low; that is, such water is a very good dielectric. The conductivity increases rapidly, however, as the water dissolves salts of various kinds; and since in the region of low concentrations, the variation of conductivity is almost directly proportional to the concentration, the conductivity may be used as a ready and fairly accurate measure of the amount of dissolved material.

Electrical-conductivity meters of several sorts are available and are used in power plants for observing the condition of the water. Some of them are arranged to read in *mhos* (i.e., 1/ohms) and others in units of an arbitrary scale.

564. Feedwater Make-Up.—(a) In the ideal power plant, operating on a closed-circuit system, the steam-generating units could be started up and kept in service indefinitely on a given supply of pure water; however, in actual plants this is impossible of attainment for many reasons. In some plants the steam exhausted from the prime movers must be wasted completely to the atmosphere; in others, part or all of it is used for process work of some sort and is wasted or contaminated; in still others it is mixed with circulating water in contact condensers and is thus lost or contaminated. Even when the steam is rejected to a surface condenser and the condensate is returned to the boiler, so that most of the working substance is carried around a closed cycle, there are some leaks of steam and water out of the system as well as leaks of gases and liquids into the system. In all actual cases, therefore, it is necessary to deal with replacement water which is more or less impure.

(b) The term **make-up water** is usually applied to the quantity of fresh or new water that must be taken into the system; and the amount of make-up supplied in a given interval of time is commonly expressed as a percentage of the weight of steam generated in the same period. Obviously, the resultant figure will be meaningless unless the period chosen is sufficiently long; hence this period is commonly 24 hours or more. The quantity of make-up required in actual plants varies from something of the order of 0.8 to 1.5 per cent in the best surface-condensing plants, and from 90 to 100 per cent in plants where most or all of the exhaust steam is wasted, except possibly some condensed in feedwater heating. It should be noted that the figure obtained for percentage of make-up may be quite deceptive because of the possibility of leaks into the system; thus, when using surface condensers, leaks of large magnitude may occur between the water side and the steam side of the condenser, the result being that the measured make-up may amount to as little as half of the total water actually entering the system. Water may also leak into the system through water-sealed glands on pumps operating under vacuum, and in other places.

565. Control of the Properties of Boiler Water.—(a) The desired properties of feedwater for steam generation include a purity such that the water (1) will not form solid deposits on boiler, or other, heating surfaces, or in the boiler feed lines; (2) will not corrode, or otherwise deteriorate, the metallic parts with which it comes in contact; and (3) will not foam or prime in the boiler; furthermore (4) the steam generated will not form deposits in the prime mover or cause corrosion therein. Very few natural waters are of such character that they meet all these requirements, and one or more of the undesirable effects enu-

merated above are generally encountered unless the properties of the water are controlled by suitable conditioning to prevent their occurrence.

(b) The **conditioning of boiler water** consists not only in treating the raw water before it enters the system, but also in continually maintaining certain chemical balances within the boiler itself. The older expression "Feedwater Treatment" is therefore somewhat less satisfactory than "Boiler Water Conditioning" or "Plant Water Conditioning," the latter referring to such control as may be necessary throughout the entire plant. In many plants, particularly the smaller industrial ones, the common practice is simply to treat the make-up chemically before or after it is mixed with the main stream; however, in the larger and better-operated plants, very exact and elaborate control methods are used. The factors that must be considered and the methods that are available are so numerous that they cannot be covered adequately in a limited space; therefore the following sections give only the more important details.

566. Boiler Compounds.—Proprietary compounds, known by this title, have been marketed under various trade names for many decades. They are intended to prevent or minimize the deposit of hard scale on boiler surfaces and the corrosion of the boiler metal; also, in some cases other advantages are claimed for them. These compounds are generally fed into the water when it is on its way to the boiler or is in a heater, and are intended, in general, to function within the boiler, bringing about the necessary changes of chemical composition as the temperature of the water rises.

Many boiler compounds undoubtedly produce good results in certain cases, in that they ameliorate the existing conditions to some extent; however, many of them are wholly worthless though innocuous, and some are positively dangerous. Since the waters dealt with are of almost every possible character and are used under many different conditions, and since it is desired to obtain definite results of a chemical nature, two things are certain: first, no one boiler compound can be applicable to all cases; and, second, unless the application of a particular compound of known composition is made after chemical study of the conditions in the particular case, its use is just as likely to make conditions worse as to make them better. The indiscriminate use of proprietary compounds is therefore most unscientific and does not conform to the standard of operating practice which is becoming more and more characteristic of the better plants. All such compounds may be analyzed to the extent required to determine the general nature of their ingredients; if the ingredients present in any one of these compounds happen to suit the conditions in a given plant, it is usually far cheaper

to buy these ingredients direct in the open market than to pay for them compounded under some trade name.

567. Water Clarification.—(a) Large quantities of **suspended matter** are often present in the water available for make-up. If this solid material enters the feedwater supply system, it will ultimately settle out by gravity in various places in which the water moves with low velocity, or else be left as solid material in the boiler when the water is converted to steam and passes off; in any event it will later have to be removed, with more or less difficulty, from some part of the system. For this reason, and also because the removal of suspended matter is desirable before the water is chemically treated, the common practice is to “clarify” the make-up before admitting it to the feedwater system. Clarification is effected by *sedimentation* or *subsidence*, the technical names for settling, and by *filtration*.

(b) **Sedimentation** consists in allowing the water to stand for a long time in a container, or in allowing it to flow through a properly baffled channel at a very low velocity. The partly clarified water is skimmed off in some convenient way from the upper part of the settling tank or chamber, and the solids that have settled to the bottom are drawn off continuously or intermittently, as may be convenient. The results obtained by sedimentation vary greatly with differences in the equipment used and with the amount and character of the suspended material. Results as low as 25 to 30 per cent removal are not uncommon, and a performance of 70 to 80 per cent is seldom exceeded. The method has the advantage, however, of cheapness and simplicity, and it may be used advantageously before filtration in order to lessen the work of the filters. Occasionally chemical changes resulting from the release of carbon dioxide, or from oxidation, occur in sedimentation tanks, and, as a result, some of the dissolved material becomes insoluble and settles out with the suspended matter.

(c) **Filtration** is generally accomplished by means of a *filter* which consists ordinarily of a bed of sand resting on gravel, the whole being enclosed in some sort of container. During operation, the water being clarified percolates downward through the filter bed and is then collected in a system of gathering pipes, or the equivalent, and led to the discharge pipe. The suspended solids are more or less completely retained by the sand which gradually becomes fouled, so that the rate of flow of the water finally decreases to a non-commercial value. The filter is then “backwashed” by forcing filtered water upward through the filter bed and allowing it to flow away at the top with the material which it has washed out of the sand bed. The best washing results are obtained when the sand is agitated during the upward flow of

water; this agitation is effected either by mechanical stirring devices or by giving the wash water so high a velocity that it makes the sand "dance."

(d) The **effectiveness of filtration** may be increased by the use of a more rapid flow of water, or by employing a *coagulant*, such as alum, which produces a flocculent precipitate when introduced into the water. This precipitate entangles the particles of suspended matter and also tends to reduce the size of the openings between the sand grains in the filter. Coagulation should be used very carefully in power-plant work and only under the guidance of one skilled in feedwater chemistry, because serious results may follow the excessive use of coagulants, or even the use of a coagulant not suited to the local conditions.

(e) **Power-plant filters** are divided into two classes, *natural or gravity-head filters*, and *pressure filters*. In the former the water flows through the bed under a slight hydrostatic head; the latter type is so arranged that the filter bed is contained within a metallic shell which can be subjected to pressure. A pressure filter is thus the equivalent of an enlargement in a water line and is frequently operated under pressures of 50 to 100 lb. per sq. in. A typical example of this type, which is the more commonly used in power plants, is shown in Fig. 958.

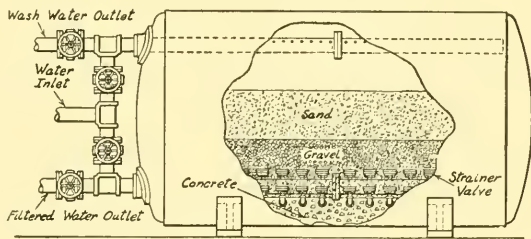


FIG. 958.—One Form of Pressure Filter.

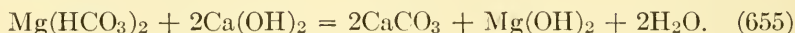
563. Water Softening by Chemical Precipitation.—(a) **Softening** consists in reducing the total hardness of water. In a much-used method, which dates back to the middle of the last century, carbonate hardness is removed by treatment with lime. The action in the case of calcium bicarbonate is merely the conversion of this *soluble* substance into the much *less soluble* mon carbonate by extraction of carbon dioxide, so that the new product may be more or less completely precipitated. The reaction is

Calcium bicarbonate + calcium hydroxide = calcium mon carbonate + water; or, in symbols,

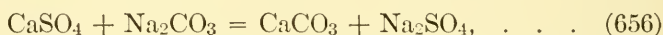


In the case of magnesium bicarbonate the reaction must be carried further because the mon carbonate is still fairly soluble. The mag-

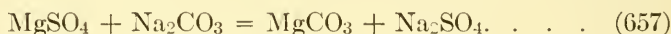
nesium hydroxide is much less soluble and is the end product sought. The reaction is



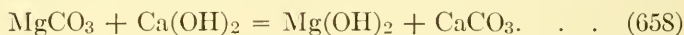
(b) The **non-carbonate hardness**, due to such salts as calcium and magnesium sulphates and chlorides, may be removed through the conversion of these salts to monocarbonates by treating the water with sodium carbonate (soda ash). The reactions are indicated in the following equations:



and



The MgCO_3 may be converted into the much less soluble $\text{Mg}(\text{OH})_2$ by treatment with lime, as indicated by the following reaction:



(c) **Comparing the treatments** just considered, that for carbonate hardness yields almost insoluble salts which can easily be removed by filtration before the water is fed to the boiler, thus preventing, to a considerable extent, the formation of hard and adherent boiler scale; and the treatment of the non-carbonate hardness merely substitutes, for the calcium and magnesium sulphates, the very soluble sodium sulphate which does not form scale. Unfortunately these treatments do not, in general, remove all the objectionable materials because the so-called insoluble salts are more or less soluble in the remaining solution.

Since bicarbonates tend to break down to monocarbonates on heating, more satisfactory results are frequently obtained by treating water with sodium carbonate at a temperature near 212 deg. fahr. The monocarbonates then present settle out to the extent that they are formed from the bicarbonates by the loss of carbon dioxide, and the sulphates are converted to carbonates by the sodium carbonate, so that with complete reactions only the very soluble sodium sulphate would remain in the water.

Methods of treatment involving the use of lime and soda ash are commonly referred to as **lime-soda processes**. When sodium carbonate is used without lime the process is generally referred to as **soda-ash treatment**. Other chemicals, such as barium carbonate and hydroxide, and sodium aluminate, are sometimes used to produce similar results.

They possess definite advantages in certain cases but are generally more costly to use. They are therefore not adopted except when conditions are such as to justify the higher cost of water treatment.

569. Commercial Chemical-Precipitation Softeners.—(a) Automatic equipment for the **continuous softening** of water by chemical methods has now come into general use in power-plant practice. Formerly the chemicals were mixed with the water in batches, at normal temperatures, and the precipitate was allowed to settle to the bottom of the reaction tank, after which the softened water was decanted from the top. Now, however, continuous dosing devices are used which feed solutions of the required chemicals at a rate proportional to the flow of water, and suitable chambers are provided in which the reactions can occur and the precipitate can settle out, and from which the softened water can be discharged at rates required to supply the demand for make-up. Of the two general types of equipment in common use, one is known as the *continuous cold-process softener* and the other as the *continuous hot-process softener*, and each is installed in connection with a filter which removes from the water such precipitate as does not settle out in the softener. The two types mentioned are essentially the same in general arrangement except that a water heater is added in the hot type. The heater is generally of an open or contact type and is located above the softener and so arranged that the water, after being heated by exhaust or bled steam, may gravitate as necessary into the softener. The chemicals are sometimes introduced into the heater and sometimes added to the water as it leaves the heater.

(b) **Hot-process softeners** have several distinct advantages. Part of the precipitation is due to the effect of heat upon the bicarbonates; thus less chemicals need be used to form the monocarbonates. Also the reactions desired take place much more rapidly at high temperatures and the resultant insoluble salts settle more rapidly and more completely. The apparatus required is therefore smaller than that for the cold process. Further, most of the gases contained in the water and those liberated by conversion of the bicarbonates to monocarbonates are vented from the heater and do not get into the feedwater system. The general arrangement of a typical hot-process plant is shown in Fig. 959.

It should be noted that while equipment of this sort can be made automatic in operation, it is still necessary to regulate the dosage in proportion to the requirements of the raw water. When the character of the water supply is uniform this is an easy matter; but when the chemical characteristics of the water change greatly, frequent analyses must be made, and the dosage regulated according to the results thereof.

570. Zeolite Softeners.—(a) **Zeolites** are naturally occurring or artificially produced silicates which possess the property known as *base exchange*. Because of this peculiarity they may be used in a water-softening process which is radically different from those described above, since it does not operate by the precipitation of insoluble salts. When zeolites, which contain sodium or other similar bases, are brought in contact with salts containing calcium or magnesium, the zeolites exchange bases with these salts; hence it follows that if water containing such salts is passed through a bed of zeolite, the latter will remove and retain the scale-forming bases, and substitute for them in the water solution the highly soluble sodium base. When the substitution of

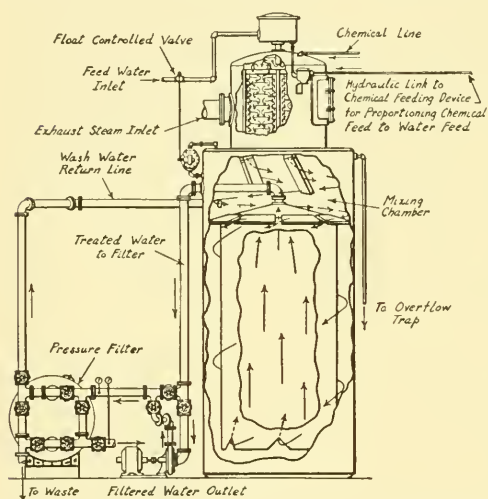


FIG. 959.—One Arrangement of a Hot-Process Softener.

bases has continued until the bed of zeolite is no longer sufficiently active, the zeolite can be regenerated by passing through it a solution of common salt. Then, the zeolite exchanges the bases it has taken from the water under treatment for the sodium of the salt solution and becomes once more ready for treating feedwater. Zeolite treatment, like lime-and-soda treatment, does not remove all the bases of the calcium and magnesium type, but, when properly handled, it does so

nearly remove them completely, that entirely satisfactory results are usually obtained in the boiler.

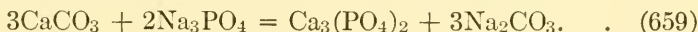
(b) The **equipment** in which zeolite softening is done is much like a sand filter, the zeolite mineral merely being substituted for sand. The water percolates slowly in one direction during softening and the salt solution in the opposite direction during regeneration. The raw water is usually filtered in advance of zeolite treatment if it contains any great amount of suspended solids, because otherwise these solids would rapidly coat the particles of zeolite and render them ineffective.

571. Comparison of Zeolite and Other Softeners.—Exact statements of general scope regarding the relative performance of the different types of chemical softeners cannot be made, as so much depends upon the

character of the water treated and on the skill and care used in control and operation. It is generally conceded that the cold lime-soda process does very well when it reduces the hardness of water to as low as 5 grains per U. S. gallon and that its commercial limit is about 3 grains. In contrast to this, the properly handled hot process should certainly reduce the hardness to 3 grains and the lower limit is generally considered as close to 1 grain. The performance of zeolite softeners, when properly handled, is seldom poorer than 1 grain of hardness per U. S. gallon, and may be even better than that. But, while the precipitation processes decrease the total quantity of soluble salts in the water, the zeolite process increases it. In the case of very hard waters treated with zeolite, the product will contain large quantities of sodium salts, and there are some conditions under which such high concentrations of the soluble salts are not desirable. Because of this, zeolites are not used to treat waters already containing large quantities of sodium salts unless the natural hardness is comparatively low.

Whenever water softening is required, the two systems must be compared and the selection should be based upon the total cost and final results obtainable with each method. The cost consideration must include not only the initial investment and the expenditures required for chemicals and attendance, but also the cost of maintenance, and that of the space occupied; cognizance should be taken also of such other factors as continuity of operation, purified water storage, and convenience.

572. Phosphate Treatment.—(a) The formation of a fine, non-adherent sludge within a boiler is preferred, of course, to the deposit of a hard scale therein. Such a substitution can be accomplished through the injection of a sodium phosphate solution directly into the boiler drum or into the feedwater supply line. This treatment is usually made supplementary to the regular chemical softening processes and, in any case, is designed to eliminate all traces of calcium carbonate from the water within the boiler. Furthermore, the addition of sodium phosphate prevents the formation of the very troublesome silicious scales which are produced under certain conditions from boiler water containing silica together with calcium or magnesium. The reaction between tri-sodium phosphate and calcium carbonate is as follows:



The formation of sodium carbonate as a by-product of this reaction may easily result in the boiler water becoming excessively alkaline, in which event further treatment is required to prevent the development of other undesirable conditions within the boiler. The calcium phosphate sludge

produced by the above reaction is easily removed when the boiler is blown down, and the formation of scale is prevented at all working pressures.

(b) The **equipment** used to inject the sodium phosphate solution into the boiler water is comparatively simple and may be arranged in several different ways. Direct injection into the boiler drum may be accomplished by means of a high-pressure plunger pump, or by using a small portion of the water discharged from the boiler feed pump to displace oil from one tank into another, and thus drive the solution out of the second tank into the boiler.³ The solution may also be introduced at any point along the feedwater supply line, on either the suction or the discharge side of the boiler feed pump; but in this case great care must be taken to prevent the formation of deposits of sodium phosphate in the various fittings through which the water flows, one satisfactory preventive being the rapid intermittent injection of the solution at the suction of the boiler feed pump.⁴

573. Feedwater Treatment by Evaporation.—When the percentage of make-up is small, the necessary quantity of water may be evaporated, and the resultant vapor condensed for use as boiler feedwater. The installation of make-up evaporators, which were considered in detail in Sect. 556, page 950, is sometimes justified when the quantity of make-up water is of the order of 2.5 per cent or less, and not over 5 per cent. This process merely transfers to the evaporator the deposit of solids which would otherwise occur in the boiler or heaters if the water were admitted directly thereto; consequently the accumulated impurities must be removed from the apparatus at intervals. In some cases the make-up water may be treated chemically before evaporation in order to decrease the amount of scale-forming material or to change its character.

574. Foaming and Priming.—(a) These terms have been used very loosely in the past and sometimes have been considered synonymous; they really refer to two distinct phenomena which, however, lead to the same final result.

Foaming properly refers to the formation of a froth or foam or collection of closely packed bubbles on the water surface from which the steam is liberated in the boiler drum. The froth may have any depth from a fraction of an inch to several inches, and in extreme cases it may actually fill the entire steam space within the drum. When foaming

³ See "Treatment of Feedwater for High-Pressure Boilers," by C. E. Joos, *Combustion*, August, 1932, pp. 19-25.

⁴ See "Phosphate Deposits Eliminated by Quick Feed," by A. B. Stickney, *Power*, June 16, 1931, p. 947.

occurs there is a tendency for the water films about the bubbles of steam to be carried over with the steam, with the result that the boiler delivers a mixture of steam and water.

Priming properly refers to a different phenomenon which may occur with or without foaming. The term covers the carrying of water into the steam space in a sort of mechanical entrainment by the steam as the result of violent ebullition, either local or general. The water carried over by priming is generally in much larger and more compact masses than it is with foaming. Priming will occur in any boiler which is driven at an excessively high rate of evaporation, although the point at which priming begins depends upon the arrangement of the boiler parts, the method of applying heat, and other variables.

(b) The **influence of the character of the water** in the boiler on both foaming and priming is not as yet clearly understood. At one time a high concentration of sodium salts was thought to be responsible; however, this alone is not sufficient to cause these phenomena in all cases. The evidence now available indicates that the underlying causes are quite complex and that the presence of finely divided solid particles is required in addition to a critical concentration. Foaming is frequently promoted by the presence, in the boiler water, of soap, resulting from the entry of sewage into the system or from the reaction of alkaline salts in the water with other impurities. The carrying over of water with the steam can frequently be prevented by a more liberal blowdown, thus decreasing the concentration with respect to both soluble and insoluble material. Foaming can generally be prevented by the use of castor oil in small quantities, or by anti-foaming compounds, most of which contain castor oil. Mechanical devices are usually included within steam drums of boilers to reduce the amount of moisture entrained in the issuing steam; several of these devices are shown in the boilers illustrated in Chapter XXXIX.

575. Control of Concentration in Boiler Water; Blowdown.—(a) The foregoing explanations of softening processes indicate that the treated water in the boiler may still contain certain **soluble salts**, or **suspended matter**, or both. During normal operation, steam is being withdrawn from the boiler and feedwater is being pumped in, and both processes are substantially continuous; hence the concentration of soluble salts and the quantity of suspended matter in the boiler water increase steadily. A high concentration of salts produces certain undesirable effects, among which is an increased tendency toward foaming and priming; the addition of more and more suspended matter results in the accumulation of sludge in certain parts of the boiler, with the possible formation of harmful sludge films on the tube surfaces.

(b) In order to control both the concentration of salts and the amount of suspended matter, the boiler is "blown down," either by an **intermittent** or a **continuous** system. Each of these systems can be arranged to give the desired control of the boiler water, as mentioned above; but when the former is used, the water from the boiler is usually discharged directly into the waste, with an accompanying loss of heat energy, whereas the latter may be installed in connection with equipment which provides for the recovery of a large portion of this heat energy.⁵

(c) The correct operation of the **intermittent system** of blowdown depends primarily upon a knowledge of the maximum permissible concentration of solids in the boiler water, and of the rapidity with which this concentration is built up. In many small, poorly operated plants the control of concentration by means of intermittent blowdown is ordinarily based entirely on guesswork. In well operated plants, both small and large, some definite method of control is used, which is based on frequent chemical analyses of both the boiler water and the feedwater; as a result of scientific study, a blowdown cycle is maintained by which the concentration of solids in the boiler water is kept below a predetermined maximum value and, at the same time, is not reduced to such an extent as to cause undue loss of heat energy, and an unnecessary increase in the quantity of make-up, through excessive blowdown.

Published reports on the operation of intermittent blowdown systems in large plants⁶ show wide variations in the maximum permissible concentration of the boiler water, and in the frequency of blowdown. The former varies from a few hundred parts per million to several thousand; the latter covers a range extending from several times per day to once in 90 days, or more. So many variables enter into the problem in any particular plant that the frequency of blowdown cannot be related directly to the allowable concentration, the amount of make-up, or the method of treatment employed; however, there exists for each plant a blowdown cycle which will provide the most favorable combination of factors from an economic standpoint.

(d) The **continuous blowdown system** has received considerable attention in this country during recent years although its development abroad began somewhat earlier. It may be used either in connection with a hot-process chemical softener, or with heat exchangers designed to transfer heat to the feedwater, or with both. In the former case the water discharged from the boiler is conducted to the softener where it gives up heat for the process, and where, at the same time, it is clarified

⁵ See Power, Nov. 25, 1930, pp. 855-857.

⁶ See Proc. N.E.L.A., 1931, p. 762.

by the removal of its suspended solids. In these installations the clarified water forms part of the make-up and is returned to the boiler, the only disadvantage being that in some cases this water still contains its soluble salts. In many instances the most favorable arrangement consists in conducting the blowdown water through suitable **blowdown heat exchangers** where it can give up heat to the make-up or the feedwater, or both, and from which it can be discharged to the sewer. One such installation⁷ is shown diagrammatically in Fig. 960, in which it should be noted that the blowdown connections in the boiler drum extend to a point just below the minimum permissible water level, this arrangement being provided in order that the boiler will not be completely drained in the event of an accident to the blowdown system. The statements which were made in (c), in regard to the control of concentration by means of an intermittent system, apply with equal force to the continuous system. From an operating standpoint, the chief difference between the two systems lies in the fact that with the former the concentration may vary over a wider range than it normally does with the latter.

(e) Since the blowdown system forms an important part of the steam-generating plant in most cases, it should be so integrated into the design of the plant that the desired economy of operation will result from the entire combination of apparatus, taking into account all factors. In general, continuous blowdown systems for use with heat exchangers are most valuable in cases where there is a scarcity of exhaust steam for heating the feedwater, although such a system may be dictated for chemical reasons alone. Also, in certain other cases, such as when extraction heaters are used, the continuous system may not show up to good advantage from a thermal standpoint. However, any such system should be capable of providing for the maintenance of a predetermined concentration of solids in the boiler water, even during fluctuations in the load, since this is the fundamental reason for the use of the blowdown.

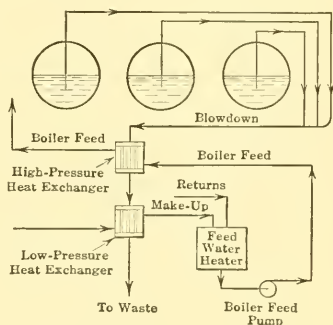


FIG. 960.—Continuous Blowdown System with High-Pressure and Low-Pressure Heat Exchangers.

576. Deaeration.—(a) Certain dissolved or mechanically entrained gases in feedwater cause considerable trouble because of their corrosive

⁷ From an article by W. J. Ryan, "The Fundamentals of Boiler Blowdown," *Power*, March 1, 1932, pp. 325-328.

action on parts of the power-plant or heating equipment with which they come in contact. The gases which produce the most undesirable effects are oxygen and carbon dioxide, the former being the more important; occasionally other gases also lead to rapid deterioration. Thus, it is often desirable to degasify the water used in the feedwater system; and since the gas to be removed has in most cases been absorbed from the air, the degasification process is known as "deaeration."

(b) **Corrosion by oxygen-bearing water** may affect any steel parts with which it comes in contact, but in most power plants the greatest difficulties are experienced within steel-tube economizers, and within the steam spaces of boilers. This is probably partly due to the liberation of

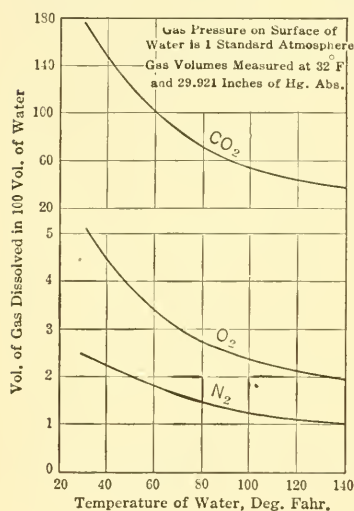


FIG. 961.—Variation of the Maximum Solubility of Gases in Water, with Changes in Temperature.

the water be agitated, it will take up a larger quantity of gas than that required to form a saturated solution under the existing conditions, the excess gas being held in mechanical entrainment from which it is gradually released when agitation ceases.

At a constant pressure, the solubility of a given gas in water decreases as the temperature is raised, the variation for three common gases being shown graphically⁸ in Fig. 961. Since the solubility at a given temperature decreases as the pressure drops, a whole family of curves could be drawn for each of the gases, and each curve of a given family would

⁸ Plotted from data by Kaye and Laby in "Physical and Chemical Constants," Longmans, Green, and Co., 1918, p. 124.

oxygen at surfaces of the metal with a rise in the temperature of the water, and partly to increased chemical activity at the higher temperatures. Water carrying carbon dioxide is also corrosive under certain conditions, although the extent to which it acts as an independent corrosive agent and to which it merely facilitates attack by oxygen are not yet understood. Oxygen in the steam also may have a detrimental effect on the parts with which it comes in contact, especially if the steam has high pressure and temperature.

(c) **The solution of a gas in water** depends upon the ability of the water to absorb this gas until a condition of saturation is reached at the existing pressure and temperature. If the

represent conditions of saturation at a specific gas pressure on the free liquid surface.

(d) **When two or more gases exist in the same space** and in contact with the free surface of water, the partial pressure of each gas determines its solubility; thus each gas acts independently in accordance with Dalton's law. Because of this fact, the water vapor in the space occupied by gas in contact with the water must always be taken into account in making measurements or calculations dealing with solubility. If a single gas were in contact with the free surface of the liquid, the pressure controlling the equilibrium of the solution would not be the total pressure measured in the space occupied by the gas, but that pressure minus the vapor pressure of the water at the existing temperature.

(e) **The difference in solubility** of the gases which compose air is responsible for the changed composition of the material released from the water during aeration, as compared with that of the air originally dissolved. If, for example, it is assumed that air consists of 20.9 per cent oxygen and 79.1 per cent nitrogen, by volume, the quantity of each component gas which will be dissolved at a given temperature can be calculated from the partial pressures of the two gases. Such a calculation, for a temperature of 60 deg. fahr. and a total pressure of one standard atmosphere, shows that the composition of the dissolved gas, if all released, would be about 33.4 per cent oxygen and 66.6 per cent nitrogen, by volume; hence there is an increased concentration with respect to the more corrosive gas. The condition is even worse when the much more soluble carbon dioxide is present in the air; for the assumptions made above, a carbon dioxide content of 0.25 per cent in the free air concentrates to such an extent as to give in solution almost 10 per cent carbon dioxide by volume.

Water does not dissolve gases to saturation instantaneously, even when violently agitated with them; nor does it release dissolved gas instantaneously when conditions are changed to bring about a lower requirement for saturation than happens to be present in the gas at the time. Therefore, the principles which have been stated must be regarded as describing the conditions that would be reached ultimately, and toward the attainment of which the system will at all times tend. In all cases the object of deaeration is to reduce the quantity of dissolved and entrained gases in water to the greatest economic extent, in order that the corrosion of the metallic surfaces in contact with the feedwater, or other hot water, or with the steam, may be properly minimized. Three *systems of deaeration* are in use; one is purely chemical, and the other two are purely physical in action.

(f) In the **chemical process** of oxygen removal, the water is passed over finely divided scrap iron or extended iron surfaces contained within a cast iron or steel shell. The oxygen of the water reacts with the iron and is thus removed from solution. The process merely localizes the corrosion in a place where it is harmless, and thus prevents, or at least minimizes, its occurrence in the more expensive equipment operating under high pressures. The apparatus sold for the removal of oxygen by this means is sometimes called a *deactivator*. This method has not been used to any extent in power plants in this country.

(g) The most common method of effecting deaeration is by means of devices known commercially as **deaerators** or **deaerating feedwater heaters**. There are many forms of these, each designed along different lines and operating, in a sense, on different principles. All, however, are expected to produce such a combination of pressures and temperatures that nearly all the gas will not only be released from solution but will also be removed as completely as possible from the equipment. Most of these deaerators heat the water to some elevated temperature and then discharge it, in sheets or spray or finely divided streams, into a vessel in which a pressure is maintained well below that corresponding to the boiling point of the water. The result is the sudden formation of vapor throughout the water at the same time that the gas is released from solution, the process being described as explosive boiling or explosive evaporation. The vapors and entrained gases are removed continuously, so that the incoming water will immediately "flash" as it enters the vessel. The evaporation of from 2 to 3 per cent of all the water entering the deaerator is generally required for satisfactory operation, the exact amount depending on the temperature and pressure relations and upon the quantity of gas to be eliminated. To conserve heat and to reduce pumping cost, the vapors given off are frequently condensed in such a way that the heat of vaporization can be returned to the system, and the non-condensable gases are removed by an air pump of some sort. Such deaerators are frequently used also as feed-water heaters, the two functions being combined in one piece of equipment. A typical example of a deaerator is illustrated in Fig. 962.

(h) The **surface condenser** is a very effective deaerator when properly built and operated. In plants where the make-up is small and is handled through evaporators, the equipment can be easily arranged so that the condenser preserves the required degree of freedom from gas. Obviously, the make-up water will contain gas as it enters the plant; if it is heated before evaporation, it will give up some but not all of this gas and the remainder will appear in the vapor from the evaporator. In the most modern plants, this vapor generally enters

the system by way of a non-contact feedwater heater, in which it is condensed. Obviously, the gas will enter the heater with the vapor; hence the heater space is drained back to the main condenser in any convenient and thermally efficient way so that the gas ultimately finds its way out of the system through the air pumps on the main condenser, as in Fig. 913.

When any method of deaeration is adopted it is necessary that a very high grade of operation be observed. The water is deaerated at the beginning of its circuit through the plant, but will pick up air very rapidly at any point of exposure; for example, a small air leak around the shaft on the suction side of the pump removing the condensate from the condenser is quite sufficient to cause serious corrosion of the high-pressure parts of the plant. Often, the feedwater in a storage tank is protected from contact with air, as by means of a floating cover or a steam blanket.

(i) The degree of degasification required to assure freedom from corrosion varies with local conditions, particularly with the pressure and therefore the temperature used in the high-pressure parts of the plant, and with the character of the water. The significance of the

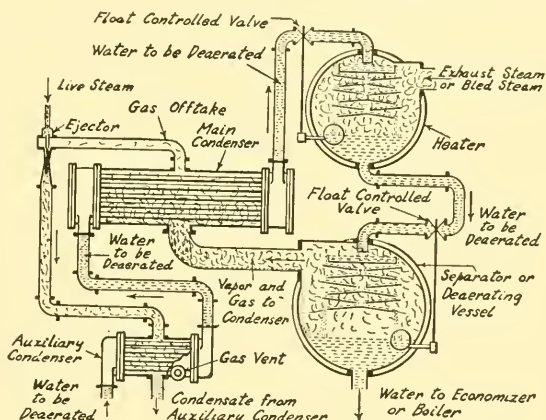


FIG. 962.—One Form of Deaerator.

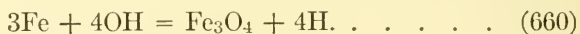
latter factor is not yet well understood, but, apparently, water which is very pure except for its gas content requires the greatest degree of gas removal, whereas water containing more salts in solution may carry larger quantities of gas without producing excessive attack on metal. It appears also, that slightly alkaline water tends to counteract the effect of oxygen in solution, to some extent. Under conditions requiring the greatest freedom from gas, the common practice is to limit the oxygen content of the water to below 0.03 cc. per liter, and removal is sometimes so nearly complete that the residual oxygen cannot be determined by present methods of test.

577. Embrittlement of Boiler Steel.—(a) The combination of excessive alkalinity and very low sulphate content in boiler water may, under certain conditions, produce an effect on the boiler metal which is

known as **caustic embrittlement**. This effect manifests itself in the cracking of the boiler plate at points below the water line, usually in riveted joints; in the cracking of rivets themselves; and in the cracking of boiler tubes at the place where they are expanded into the tube sheets. In riveted joints, the cracks frequently start at rivet holes and progress more or less radially into the plate, and sometimes they meet and develop into one crack extending between many rivet holes. In any event, the phenomenon is one which produces very undesirable results; consequently a large amount of attention has been given to the problem of eliminating this destructive action.⁹

(b) **Embrittlement cracks** are produced when metal which has previously been stressed about to its *yield point* is immersed in boiler water carrying an unusually large amount of sodium carbonate as compared to sodium sulphate. During the fabrication of the boiler, "cold work" is done on the metal adjacent to rivet holes, on the rivets themselves, and on the metal near the ends of the tubes; consequently these regions have always been highly stressed, and in this condition are much more readily attacked than are other parts of the boiler. The combination of substances which causes the attack on the metal may be present in natural waters, in water which has been given an excessive soda-ash treatment, or in water which is low in sulphates and which has been given a zeolite treatment. The presence of an adequate amount of sulphate appears to inhibit embrittlement; and other materials, such as phosphates and tannates, have also been found effective.

(c) The **mechanism of embrittlement** is not entirely understood, but the chief cause of the action seems to be the increased chemical activity of the grain boundaries in the highly stressed portions of the metallic structure, under the influence of excessive caustic alkalinity. Under working conditions, the sodium carbonate reacts with water to form sodium hydroxide which, in turn, becomes concentrated in the very narrow, or even capillary, spaces between the plates of the riveted joints, and between the plates and the rivet heads. This concentration becomes sufficient to produce a solution having an e.m.f. with respect to steel which favors the following reaction (indicated by the hydroxyl ion only):



The hydrogen liberated by this reaction penetrates readily into the grain boundaries of the cold-worked metal, and this penetration is followed by the progressive corrosion of the surfaces of the exposed grains; further-

⁹ See "Embrittlement in Boilers," by F. G. Straub, Univ. of Ill. Bulletin 216 (1930), and many other references given therein.

more, this corrosion assists in the maintenance of electrochemical activity at the tip of the advancing crack and thus favors the continued formation of hydrogen. The function of the sodium sulphate, or other inhibitor, seems to be to crystallize on the surface of the metal and lower the e.m.f. to a point at which electrochemical activity cannot be maintained. The embrittling action seems to be more rapid the higher the pressure and temperature of the steam.

(d) The **effect of caustic attack** is the production of intercrystalline cracks; that is, cracks which follow the grain boundaries within the metallic structure; and the failure of the affected part resembles that of a brittle material. However, the grains themselves have not had their physical properties altered by the attack, and a test specimen taken from the embrittled portion of a boiler will show about the same ductility as did the original metal. Owing to the fact that the seepage of sodium hydroxide through the narrow passages between the parts of a riveted joint results in the highest concentration of this compound at a point farthest away from the body of water in the boiler, embrittlement failures usually occur on the so-called dry side of the plate.

(e) Although there is still much uncertainty about the details of the chemical balance required to inhibit caustic embrittlement under all conditions of concentration, and of temperature and pressure, the A.S.M.E. Boiler Code Committee deemed it advisable to recommend the maintenance of not less than a certain **sulphate-carbonate ratio** in the boiler water. This recommendation is quoted in Table LXXX, the figures in the last column being the minimum values considered as affording safe results; and probably even higher concentrations of sulphate may be required in the case of pressures above 500 lb. per sq. in.

TABLE LXXX

RECOMMENDED RELATIONS BETWEEN SODIUM CARBONATE ALKALINITY AND SODIUM SULPHATE

Working Pressure of Boiler, Lb. per Sq. In.	Relation of		
	Sodium Carbonate Alkalinity	to	Sodium Sulphate
0 to 150	1	to	1
150 to 250	1	to	2
250 and over	1	to	3

NOTE: Sodium carbonate alkalinity is obtained by determining the amount of acid required to neutralize the water sample, using methyl orange indicator, and then calculating the weight of sodium carbonate which this quantity of acid could neutralize.

578. Summary.—The material presented in this chapter, although strictly limited in its scope, serves to emphasize the complexity of the

TABLE LXXXI
COMMON IMPURITIES IN WATER AND THE POSSIBLE EFFECT WHEN USED FOR BOILER-FEED PURPOSES

Constituent	Chemical Formula	Principal Source of Contaminating Substance	Possible Effect When Present in Boiler Waters	Treatment for Removal from Water
Suspended solids.....		Surface drainage, trade waste	Priming, foaming, sludge or scale	Plain subsidence, coagulation, filtration, evaporators
Silica.....	SiO ₂	Mineral deposits	Scale	Plain subsidence, coagulation, filtration, evaporators
Calcium carbonate.....	CaCO ₃	Mineral deposits	Scale	Softening by chemical, by base-exchange material, by evaporators
Calcium bicarbonate.....	Ca(HCO ₃) ₂	Mineral deposits	Scale	Softening by heaters, by chemical or by base-exchange materials, by evaporators
Calcium sulphate.....	CaSO ₄	Mineral deposits	Scale, corrosion	Softening by chemical, by base-exchange materials, by evaporators
Calcium chloride.....	CaCl ₂	Mineral deposits	Scale	Softening by chemical, by base-exchange materials, by evaporators
Magnesium carbonate.....	MgCO ₃	Mineral deposits	Scale	Softening by chemical, by base-exchange materials, by evaporators
Magnesium bicarbonate.....	Mg(HCO ₃) ₂	Mineral deposits	Scale	Softening by chemical, by base-exchange materials, by evaporators
Magnesium chloride.....	MgCl ₂	Mineral deposits	Scale, corrosion	Softening by chemical, by base-exchange materials, by evaporators
Free acids.....	HCl, H ₂ SO ₄	Mine drainage, trade waste	Corrosion	Neutralizing acid, followed at times by softening or evaporation
Sodium chloride.....	NaCl	Sewage, trade waste, mineral deposits	Inert but may be corrosive under some conditions	Evaporator
Sodium carbonate.....	Na ₂ CO ₃	Mineral deposits	Priming and foaming, embrittlement	Evaporator
Sodium bicarbonate.....	NaHCO ₃	Mineral deposits	Priming and foaming, embrittlement	Evaporator
Carbonic acid.....	H ₂ CO ₃	Absorption from the atmosphere, mineral deposits, decomposition of organic matter	Corrosion	Deaerator
Oxygen.....	O ₂	Absorption from the atmosphere	Corrosion	Deaerator
Grease and oil.....		Trade waste	Corrosion, deposits, priming, foaming	Coagulation, filter, evaporator
Organic matter and sewage.....		Domestic and trade wastes	Corrosion, deposits, priming, foaming	Coagulation, filter, evaporator

problems involved in controlling the properties of boiler feedwater. The rule-of-thumb practices formerly used in feedwater treatment have given way to accurate chemical analysis and scientific methods of control, both before and after the water enters the boiler. Although great advances have been made in this field in recent years, the causes underlying some of the undesirable effects produced by feedwater have not yet been determined; hence much work remains to be done, especially in improving the properties of water intended for use in boilers operating at the high pressures which are now being introduced in central-station practice. Fortunately for the student, a large amount of information has been, and is being, published on the general subject of boiler-water control; among these may be mentioned the serial reports and bibliographies on the treatment of feedwater formerly issued by the Prime Movers Committee of the National Electric Light Association, A.S.M.E. reports, and the annual summary of feedwater treatment published in the last issue of *Power Plant Engineering* for each calendar year.

A convenient summary of the common impurities found in water, together with their sources and effects, and the means of controlling them, is given ¹⁰ in Table LXXXI.

¹⁰ From "Natural Waters and Their Impurities," by S. T. Powell, in *Power*, July 6, 1926, p. 13, and also given in "Boiler Feedwater Purification," by the same author and published by McGraw-Hill Book Co. (1927).

CHAPTER XLVII

PROPERTIES OF MIXTURES OF AIR AND WATER VAPOR

579. General.—In some branches of engineering, fluids that are mixtures of gases and vapors are in constant use. For example, the atmospheric air used for combustion, for ventilation and heating, for drying, and for water cooling is normally a mixture of dry air and water vapor; also, the products of combustion from furnaces and internal-combustion engines contain considerable amounts of water vapor, and the exhaust from steam prime movers is a mixture of steam and gases. Hence, a knowledge of the methods of determining the properties of gas-vapor mixtures is important to the engineer. Only the properties of mixtures of dry air and water vapor will be herein discussed, but many of the fundamental principles may be applied to mixtures of other gases and other vapors.

580. Principles and Definitions.—(a) A mixture of chemically inert gases and vapors, or a mixture of vapors alone, tends to become homogeneous through diffusion, and **Dalton's Law of Partial Pressures**¹ applies, just as when the mixture is composed only of gases. In fact, the key to the simplest treatment of a mixture of air and water vapor is to *treat the water vapor (whether saturated or superheated) as a gas* with a gas constant, R , of 85.6 ft.-lb. per deg. fahr. abs., and with a constant pressure specific heat, c_p , of 0.45 B.t.u. per lb. per deg. fahr. That these values are quite exact for the usual low partial pressures of the vapor and for temperatures below 400 deg. fahr. is evident from inspection of Fig. 963, in which the best available values of the gas constant, R , and of the heat content, h , of saturated steam and of superheated steam for pressures of 1 and 2 lb. per sq. in. abs. are shown plotted against temperature. The data for the points shown on the graph are based upon Keenan's "Steam Tables," and smooth curves are drawn through these points.

(b) The **water vapor** in the mixture may be either *saturated* or *superheated*. If the vapor is in the dry saturated condition, the mixture is commonly said to be "saturated," since there could be no further addition to the vapor content without consequent condensation and the

¹ See the discussion of gas mixtures in Sect. 75, Part I.

appearance of water. The presence of water in such a mixture does not affect the amount or condition of the vapor. Such mixtures of dry gas, vapor, and liquid exist not only at times in the atmosphere, but also in steam condensers and feedwater heaters, in the intakes of their air pumps or ejectors, in parts of air-washing and conditioning apparatus, in the discharge from carburetors, and in fuel mixtures undergoing compression in internal-combustion engines using liquid fuel. However, the gas-vapor mixture most commonly existent is that mixture containing *superheated* vapor, such as the usual atmosphere in which the weight of water vapor present is not sufficient to cause saturation at the existing temperature. In order to determine the exact state of the water vapor in a mixture, the partial pressure of the vapor must be found.

(c) The common engineering method of determining this actual partial pressure of the water vapor is the **psychrometric, or evaporative**. In this method, the dry-bulb and wet-bulb temperatures and the total pressure of the mixture are determined.

The sling psychrometer is used to find the dry- and wet-bulb temperatures of the mixture. This instrument consists of dry- and wet-bulb thermometers suitably mounted and provided with a handle by means of which they may be rapidly whirled through the air. The bulb of the dry-bulb thermometer is uncovered, while that of the wet-bulb thermometer is encased in a fine-mesh fabric bag kept moistened with clean water. If the water vapor in the air is in the dry saturated condition (i.e., if the air is "saturated") both thermometers will give the same reading. If the water vapor is superheated, the wet-bulb will be lower than the dry-bulb temperature, and this difference is called the *wet-bulb depression*. The reason for this will now be investigated.

Experiments show that the wet-bulb will attain a steady temperature for any given condition of the atmosphere. This indicates that a dynamic equilibrium exists between the rate at which heat is transferred from the air to the water and the rate at which this heat is utilized in evaporating water from the rapidly moving bulb. The rate at which

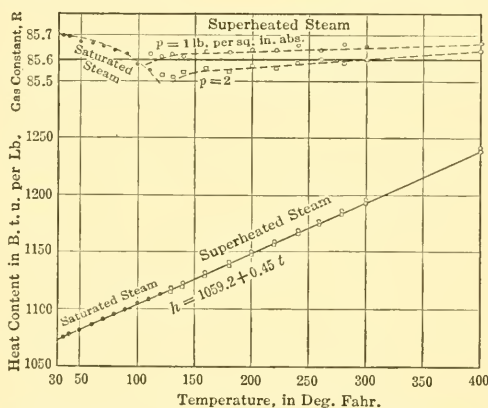


FIG. 963.—Some Properties of Low-Pressure Steam at Temperatures Below 400 Deg. Fahr.

heat is transferred from the air to the water is substantially proportional to the film coefficient of heat transfer and to the wet-bulb depression.² The rate of heat utilization by the evaporation of water is proportional to the coefficient of vapor diffusion, the latent heat of vaporization, and the difference between the saturation pressure at the water temperature and the actual partial pressure of the water vapor in the air. Therefore the wet bulb gives a steady temperature only when there is a balance between the rates of heat transfer and of heat utilization. For example, suppose the temperature of the water on the wet bulb be initially very high; the corresponding saturation pressure is high, the rate of heat removal from the bulb by evaporation exceeds the rate of heat transfer from the air to the water and consequently the temperature of the bulb diminishes. On the other hand, if the temperature of the water on the bulb is very low, the rate of heat transfer to the bulb from the air exceeds the rate of heat utilization by evaporation and the bulb temperature rises, approaching an equilibrium value.

The **Ferrel psychrometric formula**,³ rationally based and with empirical constants that hold for temperatures between -40 and 140 deg. fahr., expresses this dynamic equilibrium with sling psychrometers as follows:

$$\frac{p' - p}{t - t'} = 0.000367 p_m \left(\frac{t' + 1539}{1571} \right), \quad . \quad . \quad . \quad (661)$$

where p = the actual partial pressure of the water vapor in the mixture, in in. Hg abs.

p' = the saturation pressure at the wet-bulb temperature, in in. Hg abs.

t = the dry-bulb temperature, in deg. fahr.

t' = the wet-bulb temperature, in deg. fahr.

p_m = the total pressure of the mixture (commonly the barometric pressure), in in. Hg abs.

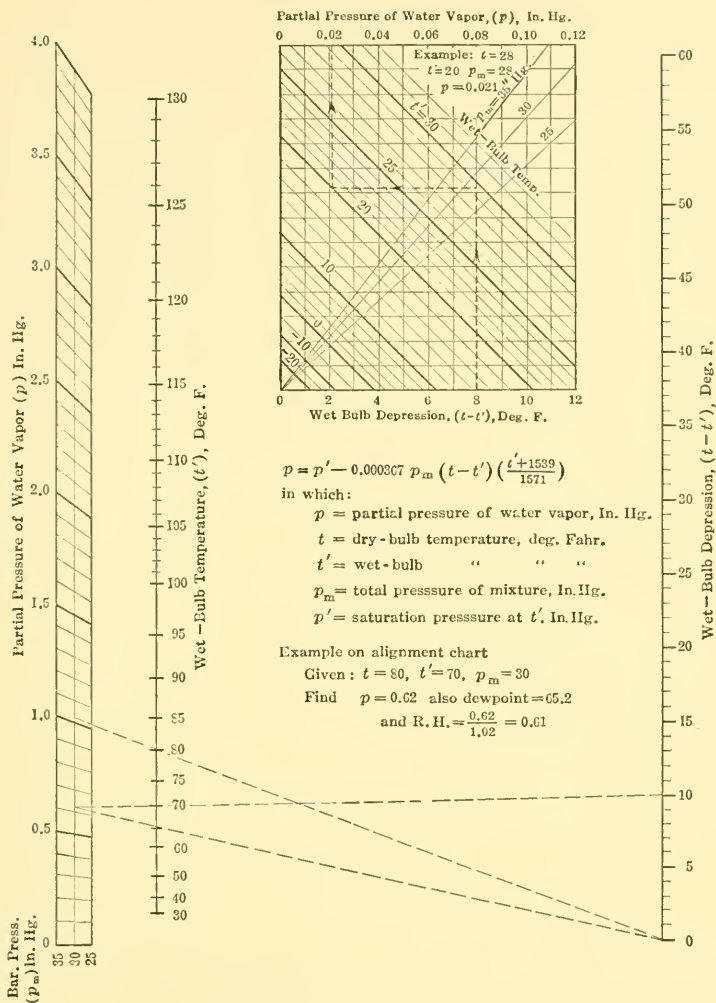
With the dry-bulb and wet-bulb temperatures and the total mixture pressure experimentally determined, the actual partial pressure of the water vapor in the mixture may be found from Eq. (661). An alignment chart solution ⁴ of this equation is shown in Fig. 964.

The saturation pressure of water vapor for temperatures at 32 deg. fahr. and above may be found from steam tables or charts. For con-

² Heat transfer by radiation is relatively negligible so long as the velocity of the whirling bulb is high, say above 1000 ft. per min. The stationary hygrometer is unsatisfactory for this determination because of the uncertain and varying effect of heat transfer by radiation.

³ U. S. Dept. of Agriculture, Weather Bureau Bull. 235.

⁴ See "A Review of Psychrometric Charts," by C. O. Mackey, Heating and Ventilating, August, 1931, p. 59.



venience, Table LXXXII is included, in which the **saturation pressures of water vapor in contact with ice** at temperatures from -20 to 31 deg. fahr. are given. The values in this table are from the Weather Bureau Bulletin 235.

(d) With the partial pressure of the water vapor in the mixture known, the **dew point** can then be found, since the dew point of the mixture is the saturation temperature corresponding to this vapor pressure. If the mixture were cooled at constant pressure, the dew

TABLE LXXXII

PRESSURES OF SATURATED WATER VAPOR IN CONTACT WITH ICE

t Deg. Fahr.	p In. Hg	t Deg. Fahr.	p In. Hg	t Deg. Fahr.	p In. Hg	t Deg. Fahr.	p In. Hg
-20	0.0126	-7	0.0260	6	0.0515	19	0.0979
-19	0.0133	-6	0.0275	7	0.0542	20	0.103
-18	0.0141	-5	0.0291	8	0.0570	21	0.108
-17	0.0150	-4	0.0307	9	0.0600	22	0.113
-16	0.0159	-3	0.0325	10	0.0631	23	0.118
-15	0.0168	-2	0.0344	11	0.0665	24	0.124
-14	0.0178	-1	0.0363	12	0.0699	25	0.130
-13	0.0188	0	0.0383	13	0.0735	26	0.136
-12	0.0199	+1	0.0403	14	0.0772	27	0.143
-11	0.0210	2	0.0423	15	0.0810	28	0.150
-10	0.0222	3	0.0444	16	0.0850	29	0.157
- 9	0.0234	4	0.0467	17	0.0891	30	0.164
- 8	0.0247	5	0.0491	18	0.0933	31	0.172

point is that temperature at which condensation of the water vapor would start. The dew point is shown as t_3 in Fig. 965. The relationship between dry-bulb temperature, wet-bulb temperature, and dew

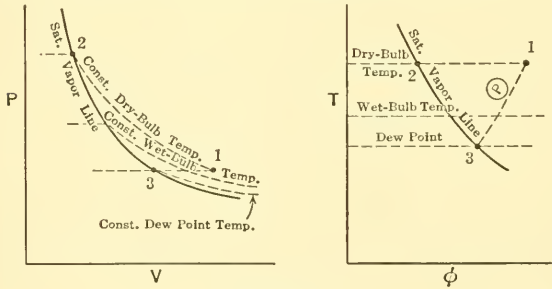


FIG. 965.

point is also shown in the same figure for the mixture in which the water vapor is superheated.

(e) The term **humidity** is used to designate the amount of water vapor mixed with dry air in the atmosphere.

Relative humidity may be defined *either* as the ratio of the actual *density* of the water vapor in the mixture to the density of saturated vapor at the dry-bulb temperature of the mixture, *or* as the ratio of the actual *partial pressure* of the vapor to the pressure of saturated vapor at the dry-bulb temperature.

Absolute humidity, as used in this book and by certain other authors, means the weight of water vapor mixed with a unit *weight* of dry air, although this term is frequently used to denote the weight of water

vapor in unit *volume* of the mixture. With this latter meaning, absolute humidity becomes nothing but the actual density of the water vapor in the mixture, and a second name for this property seems superfluous and confusing.

(f) The **barometric pressure**, or the total pressure of the atmospheric mixture of dry air and water vapor, must be found experimentally, as by means of a barometer. This pressure depends upon climatic conditions and upon altitude. In many problems in heating and ventilating, air conditioning, drying, and water cooling, the variation in barometric pressure is ignored and a standard value of 29.92 in. Hg used. On the other hand, the change in barometric pressure with altitude has an important effect upon the performance of chimneys, fans, furnaces, internal-combustion engines, air compressors, and other apparatus, since the density of dry air depends upon this pressure and upon the temperature.

For evaluating airplane performance, calibrating aeronautical instruments, and for other purposes, a **standard atmosphere**⁵ for the United States has been adopted by the Bureau of Standards, Weather Bureau, National Advisory Committee for Aeronautics, Army Air Corps, and the Navy Bureau of Aeronautics. The effect of altitude on the pressure, temperature, and density of *dry air* for the assumed standard atmosphere is shown in Fig. 966.

At sea level, the standard pressure is 29.921 in. Hg, the standard temperature, 59 deg. fahr., and the standard density, 0.0765 lb. per cu. ft. The "isothermal level" is assumed to be at an altitude of 35,332 ft. Above this level, the standard temperature of the atmosphere is assumed to remain constant at -67 deg. fahr.

581. Properties of Mixtures of Air and Water Vapor.—(a) The two properties of a mixture of dry air and steam that are necessary in the applications of the principles of conservation of mass and conservation of energy to processes in which such mixtures are the working substances are the absolute humidity and the heat content or enthalpy. It is convenient to determine these properties *per pound of dry air* with its associated steam, because, in many processes of change of state, the

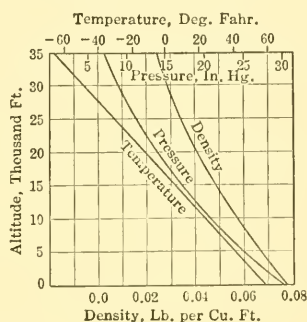


FIG. 966.—The Standard Pressure, Temperature, and Density of Dry Air at Different Altitudes.

⁵ See Publication 82, U. S. Bureau of Standards; also Technical Reports 147, 218, and 246 of the National Advisory Committee for Aeronautics.

weight of dry air undergoing the process remains unchanged while the weight of water vapor accompanying this air may be changed by condensation or by evaporation.

(b) The **absolute humidity** may be found either with or without the aid of steam charts or tables after the dry-bulb temperature of the mixture and the partial pressure of the water vapor are known. Using steam charts, the specific volume of the steam in the mixture may be located from the known degree of superheat (dry-bulb minus dew point), and pressure (p). The density of the steam may be found from the reciprocal of the specific volume at that state. Or using steam tables, the density of saturated steam at the dry-bulb temperature may be read, and the actual density of the steam in the mixture is the product of this tabular value and the relative humidity. In either case, the absolute humidity is then equal to the ratio of the density of the steam in the mixture to the density of the dry air, since each constituent of the mixture occupies the same volume.

An alternative and simpler method of determining absolute humidity is based on the assumption previously mentioned, that the steam obeys the gas laws with a gas constant, R , of 85.6. Then the absolute humidity is

$$w' = \frac{\frac{p}{85.6T}}{(p_m - p)} = 0.622 \frac{p}{p_m - p}, \quad . \quad . \quad . \quad (662)$$

where w' = the absolute humidity, in lb. per lb. of dry air.

p = the partial pressure of the steam.

p_m = the total pressure of the mixture, in the same units as p .

(c) The **heat content**, or **enthalpy**, of the mixture is the sum of the heat content of the dry air and the heat content of the associated water vapor. The heat content of steam and water is referred to the saturated liquid at 32 deg. fahr. as a datum, so that tabular or charted values of the heat content of the steam and water may be used. A lower datum temperature is desirable for the heat content of the dry air in order to avoid the use of negative values of heat content at temperatures below 32 deg. fahr., and a datum of -40 deg. fahr. will be used.⁶ The heat content of a mixture of dry air and water vapor is

$$h' = 0.24(t + 40) + w'h_v, \quad . \quad . \quad . \quad (663a)$$

⁶ If any water is supplied or abstracted during any process of change of state of a mixture, or if the heat content of a mixture of dry gas, steam, and water is to be found, the heat content of the water must be referred to the usual datum.

where h' = the heat content of the mixture, in B.t.u. per lb. of dry air.

t = the dry-bulb temperature, in deg. fahr.

w' = the absolute humidity, in lb. per lb. of dry air.

h_v = the heat content of the water vapor in B.t.u. per lb.

The heat content of the vapor, h_v , may be found directly from tables or charts if the partial pressure and temperature of the steam are known. However, to avoid the use of steam tables and charts, an empirical equation for the heat content of the steam, shown graphically in Fig. 963, may be used. Then, Eq. (663a) becomes

$$h' = 0.24 (t + 40) + w' (1059.2 + 0.45t). \quad . \quad . \quad (663b)$$

(d) The **volume** occupied by a mixture of dry air and steam is another property that is necessary in many calculations, particularly in determining the duty of fans handling such mixtures. Assuming that the steam obeys the gas laws, one may readily prove that the volume occupied by the mixture *per pound of dry air* present, is, in cubic feet,

$$V' = \frac{53.3T}{70.7(p_m - p)} = \frac{85.6w'T}{70.7p}. \quad . \quad . \quad . \quad . \quad (664)$$

The volume occupied by the mixture *per pound of the mixture*, or the true specific volume of the mixture is, in cubic feet,

$$\bar{V} = \frac{(53.3 + 85.6w')T}{70.7p_m(1 + w')} = \frac{1}{1 + w'}V'. \quad . \quad . \quad . \quad . \quad (665)$$

(e) Other properties of the mixture, such as internal energy and entropy, may be evaluated when necessary by adding the specific property (internal energy or entropy) of the dry air to the corresponding specific property of the associated water vapor.

582. Processes of Mixtures of Air and Water Vapor.—(a) The **energy equations** of Chapter IV, Part I, such as the work, simple energy, specific heat, and general energy equations, may be applied to processes of air-water vapor mixtures when the respective hypothetical conditions of application are met.

A general energy equation, that is particularly useful for analyzing many processes of heating and cooling that take place in air-conditioning units, driers, and water-cooling devices such as cooling towers, may now be developed. Consider the rectangle in Fig. 967 to represent any apparatus through which a mixture of air and water vapor is passing. By a proper choice of the directions of the arrows or by the addition of more energy arrows, or both, the rectangle may represent a humidifier, a drier, a cooling tower, or the like. In general, a mixture of 1 lb. of air

and associated water vapor enters the apparatus in state 1 and leaves in state 2. In the case shown, water is supplied to the apparatus in state 3. The general energy equation applied to this case, in which there is no mechanical energy delivered to or by the apparatus, between the sections 1 and 2, and in which the fluids undergo no appreciable change in velocity or elevation by virtue of their passage through the apparatus, becomes

$$h'_1 + E_h + (w'_2 - w'_1) h_{l_3} = h'_2 + R_c, \quad . \quad . \quad . \quad (666a)$$

or

$$E_h = h'_2 - h'_1 - (w'_2 - w'_1) h_{l_3} + R_c, \quad . \quad . \quad . \quad (666b)$$

where E_h = the total quantity of heat supplied under the control of the operator, in B.t.u. per lb. of dry air passing through the apparatus.

h'_1 = the heat content of the mixture entering the apparatus, in B.t.u. per lb. of dry air.

h'_2 = the heat content of the mixture leaving the apparatus, in B.t.u. per lb. of dry air.

w'_1 = the absolute humidity at entrance, in lb. per lb. of dry air.

w'_2 = the absolute humidity at exit, in lb. per lb. of dry air.

h_{l_3} = the heat content of the water supplied, in B.t.u. per lb.

R_c = the quantity of heat lost externally from the apparatus (owing to imperfect insulation), in B.t.u. per lb. of dry air.

If a fan were placed between sections 1 and 2 to produce a rapid flow of the mixture, the amount of energy supplied to operate this fan should properly be included in the energy equation; usually, however, the omission of this energy will not cause a serious error, because its magnitude may be small when compared with the other energy terms.

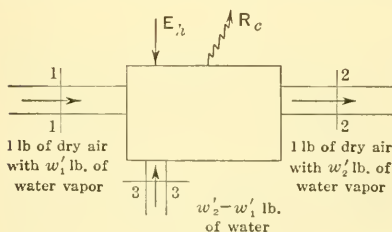


FIG. 967.

(b) The **adiabatic saturation** or **adiabatic cooling process** may be studied for the purpose of developing certain psychrometric principles of use in analyzing air-conditioning, water-cooling, and drying operations. The meaning of this process may be explained with the aid of Eq. (666a) and Fig. 968, in which the heavy lines represent insulation. Consider a mixture of dry air and water vapor, initially unsaturated and in state 1, flowing, with a velocity high enough to insure vapor diffusion, in contact with an extensive surface of water through a tunnel and leaving in state 2. No external heat transfers are taking place; the only heat exchange is between the air and water, and only in this sense can the process be called "adiabatic." Experiments conducted

by W. H. Carrier and D. C. Lindsay ⁷ indicate that, in such a process with sufficient water surface, the exit mixture will be saturated at the *initial wet-bulb temperature of the mixture*. Also W. K. Lewis ⁸ has demonstrated, mathematically, that if the water supplied is at the initial wet-bulb temperature of the mixture and if the amount of vapor carried by the air is small, the temperature of the water and air reach an end-point value that is the same as the initial wet-bulb temperature of the mixture. In short, in the process of adiabatic saturation, the dry-bulb temperature of the mixture drops to the initial wet-bulb temperature as a limit, the weight of vapor mixed with each pound of dry air increases to that required for saturation at the wet-bulb temperature as a limit, and the wet-bulb temperature remains substantially constant.

Applying Eq. (666a) to the adiabatic saturation process, assuming the wet-bulb temperature to remain constant, and noting that $E_h = 0$ and $R_c = 0$, also that the final dry-bulb temperature of the mixture, t_2 ,

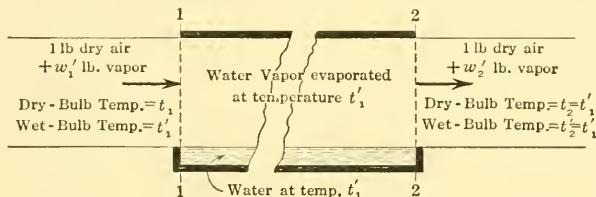


FIG. 968.—A Diagram of the Adiabatic Saturation Process.

and the temperature of the supply water are both equal to the initial wet-bulb temperature, t' , then, from Fig. 968, it follows that:

$$h'_1 + 0 + (w'_{s'} - w') h_{v'} = h'_2 + 0. \quad (667a)$$

By substituting for h'_1 and h'_2 from Eq. (663a), it follows that

$$w' = \frac{w'_{s'} r' - 0.24 (t - t')}{h_{v_1} - h_{v'}}, \quad (667b)$$

where $w' =$ the weight of water vapor mixed with each lb. of dry air at any dry-bulb temperature, t , and corresponding wet-bulb temperature, t' , in lb.

$w'_{s'} =$ the weight of saturated water vapor mixed with each lb. of dry air for saturation at the wet-bulb temperature, in lb.

$r' =$ the latent heat of vaporization at the wet-bulb temperature, in B.t.u. per lb.

$h_{v_1} =$ the heat content of the water vapor in the mixture at the dry-bulb temperature, t , in B.t.u. per lb.

$h_{v'} =$ the heat content of the liquid at the wet-bulb temperature, t' , in B.t.u. per lb.

⁷ "The Temperature of Evaporation of Water into Air," Trans. A.S.M.E., Vol. 46, 1924, p. 739.

⁸ "The Evaporation of a Liquid Into a Gas," Trans. A.S.M.E., Vol. 44, 1922, p. 325.

Since $h_{v_1} = 1059.2 + 0.45t$, and since the latent heat of vaporization at the wet-bulb temperature is $r' = 1059.2 + 0.45t' - h_{v'}^u$, then

$$w' = \frac{w'_s r' - 0.24 (t - t')}{r' + 0.45 (t - t')} \quad (668)$$

With the barometric pressure and the dry-bulb and wet-bulb temperatures known, this equation gives a method of finding the weight of vapor mixed with each pound of dry air (and the partial pressure of that vapor) that is an alternative to the use of the empirical Ferrel formula of Eq. (661) and the rational equation, Eq. (662). Since the application of the Ferrel formula is limited to temperatures below 140 deg. fahr., and Eq. (668) is not limited to this temperature field, the latter equation is more general.

As an **illustration** of the two methods, consider the following example:

Find the partial pressure and the weight of water vapor mixed with each pound of dry air for a dry-bulb temperature of 90 deg. fahr., a wet-bulb temperature of 70 deg. fahr., and a barometric pressure of 28 in. Hg.

Method 1.—Using Eq. (661), the partial pressure of the vapor is

$$\begin{aligned} p &= 0.74 - 0.000367 \times 28 \times 20 \times \frac{1609}{1571} \\ &= 0.74 - 0.21 = 0.53 \text{ in. Hg.} \end{aligned}$$

Using Eq. (662), the weight of vapor is

$$w' = 0.622 \frac{0.53}{27.47} = 0.012 \text{ lb. per lb. of dry air.}$$

Method 2.—Using Eq. (668), the weight of vapor is

$$w' = \frac{1052.7 w'_{s'} - 0.24 \times 20}{1052.7 + 0.45 \times 20}.$$

From Eq. (662)

$$w'_{s'} = \frac{0.622 \times 0.74}{27.26} = 0.01688.$$

Hence

$$w' = 0.0122 \text{ lb. per lb. of dry air.}$$

Then, by using Eq. (662) again, the partial pressure of the vapor is

$$p = \frac{0.0122 \times 28}{0.6342} = 0.538 \text{ in. Hg.}$$

583. Psychrometric Charts.—(a) Many psychrometric charts, or graphical solutions of the preceding psychrometric equations,⁹ have

⁹ For illustrations, and references to the original charts, see "A Review of Psychrometric Charts," by C. O. Mackey, Heating and Ventilating, June, July, and August, 1931.

been constructed to reduce the labor of solving problems in air conditioning, drying, and water cooling. Most of these charts are drawn for one constant value of the total pressure of the mixture (usually 29.92 in. Hg abs.), and are satisfactory for the solution of many problems since the effect of normal deviations from this barometric pressure is small.

A new form of psychrometric chart that is easy to construct and shows the principal properties of mixtures of dry air and steam necessary in applying the conservation of mass and conservation of energy principles is shown ¹⁰ in Fig. 969. This chart covers a range of dry-bulb temperatures from 32 deg. fahr. to 130 deg. fahr., a range of wet-bulb temperatures from 32 deg. fahr. to 100 deg. fahr., and a range of total mixture pressures from 25 in. Hg to 30 in. Hg. Similar charts may of course be drawn to cover other ranges of temperature and pressure.

On this chart, the heat content of the mixture is plotted as ordinate against the dry-bulb temperature as abscissa. Lines of constant absolute humidity, w' , are drawn on this coordinate system by the aid of Eq.

(663*b*). The absolute humidities and the heat contents of mixture having a relative humidity of 100 per cent are then calculated and plotted against dry-bulb temperature; the position of these curved lines depends upon the total or barometric pressure; and such lines are shown in this figure for barometric pressures of 25 and 30 in. Hg only. Lines of constant wet-bulb temperature are then added with the aid of Eq. (668); the position of these lines depends upon the barometric pressure, and those shown by solid lines on the chart are for a barometric pressure of 30 in. Hg only. However, since these lines of constant wet-bulb temperature are nearly straight and parallel, their positions for

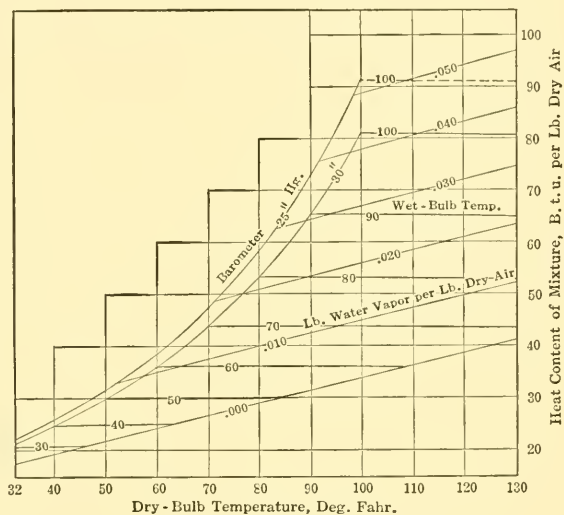


FIG. 969.—Psychrometric Chart.

¹⁰ This is merely a skeleton form of the more complete charts devised and drawn by Carl Shabtac and C. O. Mackey.

barometric pressures other than 30 in. Hg may easily be estimated by following over from the curve of 100 per cent relative humidity as indicated by the dotted line at the top of the chart for a wet-bulb temperature of 100 deg. fahr.

For example, for a dry-bulb temperature of 120 deg. fahr., a wet-bulb of 100 deg. fahr., and a total pressure of 30 in. Hg, the absolute humidity is 0.038 lb. per lb. of dry air, and the heat content of the mixture is 80.5 B.t.u. per lb. of dry air; for the same dry- and wet-bulb temperatures, but a total pressure of 27 in. Hg, the absolute humidity is 0.043 lb. per lb. of dry air, and the heat content of the mixture is 86.5 B.t.u. per lb. of dry air. The dew point may also be read from the abscissa scale by following the line of constant absolute humidity to the curve for a relative humidity of 100 per cent and the proper total pressure; for example, in the first illustration above, the dew point is 96 deg. fahr., and in the second illustration, 96.5 deg. fahr. Although the lines of constant wet-bulb temperature (except 100 deg. fahr.) do not appear for total pressures other than 30 in. Hg, their position may be estimated; e.g., for a dry-bulb temperature of 80 deg. fahr., a wet-bulb of 70 deg. fahr., and a total pressure of 27 in. Hg, the absolute humidity is 0.015 lb. per lb. of dry air, and the heat content is 45.5 B.t.u. per lb. of dry air.

(b) A property of the mixture called **total heat** appears upon many of the older psychrometric charts and in many tables. This property has not been mentioned herein, previously, because it is not essential to the fundamental development of the preceding principles. However, since this term is in common use in some text books and articles on heating, ventilating, and air conditioning, it will be explained.

The "total heat" of a mixture of dry air and water vapor, as originally defined by W. H. Carrier,¹¹ is not equal to the *heat content* or *enthalpy* of the mixture as defined in Sect. 581. Instead, the "total heat" of a mixture of dry air and water vapor (with constant values of the constant pressure specific heats of dry air and water vapor, that hold for the usual conditions of the mixture, and with a datum temperature of -40 deg. fahr. for the dry air, inserted in the defining equation) is

$$\Sigma = 0.24(t + 40) + w'[r' + 0.45(t - t')], \quad . \quad . \quad . \quad (669)$$

where Σ = the "total heat" of the mixture, in B.t.u. per lb. of dry air.

t = the dry-bulb temperature, in deg. fahr.

t' = the wet-bulb temperature, in deg. fahr.

w' = the weight of water vapor mixed with each pound of dry air, in lb.

r' = the latent heat of vaporization of water at the wet-bulb temperature, t' , in B.t.u. per lb.

Since this definition holds for any mixture of dry air and water vapor,

¹¹ Trans. A.S.M.E., 1911, p. 1039.

the "total heat" of a mixture with a relative humidity of 100 per cent and at the wet-bulb temperature, t' , is

$$\Sigma' = 0.24(t' + 40) + w'_s r', \quad . \quad . \quad . \quad . \quad (670)$$

in which Σ' = the "total heat" of a mixture of dry air and *saturated* water vapor, in B.t.u. per lb. of dry air.

w'_s = the weight of *saturated* water vapor mixed with each pound of dry air, in lb.

For the process of adiabatic saturation at constant wet-bulb temperature, Eq. (668) has been derived; by inspection, this equation (668) follows only if $\Sigma = \Sigma'$. In short, the "total heat" of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. Values of the "total heat" may be calculated for saturation at the wet-bulb temperature for a single barometric pressure from Eq. (670), and these values may be plotted against wet-bulb temperature as in the psychrometric charts, or tabulated as in tables originally prepared by the late Professor G. A. Goodenough. Then, the "total heat" of the mixture may be found if the wet-bulb temperature is known, regardless of the dry-bulb temperature or absolute humidity.

(c) It is common but inexact practice to use this so-called total heat in place of true heat content or enthalpy in applications of the conservation of energy principle. Although an approximate answer may often be quickly found by this method, the practice cannot be justified, thermodynamically.

As an illustration of the exact and the approximate methods of solution, consider the following **example**:

A mixture of dry air and water vapor with $t_1 = 90$ deg. fahr., $t'_1 = 70$ deg. fahr., and $p_m = 28$ in. Hg abs., is cooled under steady flow conditions in an air-conditioning unit to a final temperature t_2 , of 50 deg. fahr. Assuming that the condensate is removed at 50 deg. fahr., find the quantity of heat removed per pound of dry air passing through the unit.

The absolute humidity of the mixture entering the unit is, from Eq. (668), $w'_1 = 0.012$ lb. per lb. of dry air. From Eq. (662), the absolute humidity of the exit mixture is $w'_2 = 0.00815$ lb. per lb. of dry air. The weight of condensate, $w'_1 - w'_2$, is 0.00385 lb. per lb. of dry air. (This work may be checked on Fig. 969.)

Using the exact method of calculation, from Eq. (663b), $h'_1 = 44.4$, and $h'_2 = 30.4$ B.t.u. per lb. of dry air. Applying the conservation of energy principle, as in the development of Eq. (666a), the quantity of heat removed from the mixture is $(44.4 - 30.4 - 0.00385 \times 18)$ or 13.9 B.t.u. per lb. of dry air. (This may be checked with Fig. 969.)

As defined by Eq. (669), the "total heat" of the mixture entering the unit is 43.9, and the "total heat" of the exit mixture, 30.3 B.t.u. per lb. of dry air. Using "total heat" in place of heat content, the approximate answer is $(43.9 - 30.3 - 0.00385 \times 18)$ or 13.5 B.t.u. per lb. of dry air. This result is in error by only 3

per cent; if the older psychrometric charts or tables were used, since the "total heat" appearing in these tables is for $p_m = 29.92$ in. Hg, the result would be in error by about 5 per cent, not only because of using "total heat" in place of heat content, but also because the actual total pressure of the mixture does not happen to be the standard barometric pressure.

Evidently, then, the use of "total heat" in place of heat content may not introduce an extremely serious error, but for the sake of consistent thermodynamics, "total heat" should disappear from the jargon of heating and ventilating, especially, since graphical charts such as Fig. 969 may be employed to simplify the solution of numerical problems.

CHAPTER XLVIII

WATER COOLING WITH ATMOSPHERIC AIR

584. General.—The supply of cold water available for circulation through a condenser (for condensing steam, ammonia, or other vapors), through the jackets of an internal-combustion engine, or through oil coolers or other similar heat-exchange apparatus, is often unsatisfactory, limited in amount, or costly. The water may be unsatisfactory on account of contamination with sewage or acids, or for other reasons. When the water is scarce or expensive, it should not be wasted after a single passage through the apparatus, as is done when using a plentiful supply of cheap water; instead, a limited amount of suitable water may be used repeatedly, provided that it can be cooled between stages of use. The temperature range through which the circulating water must be cooled is, of course, equal to that through which the water is heated in the apparatus served. Atmospheric air may be used as the cooling medium in three different types of cooling systems: cooling ponds, spray ponds, and cooling towers.

585. Heat Transfer in Water Cooling.—(a) When the temperature of a water surface is *higher than the dry-bulb temperature* of the surrounding air, heat is transferred from the water to the air by radiation and convection, and is also removed from the water through evaporation into the ambient air. The rate of such heat transfer by radiation is extremely slow, because of the low absolute temperatures and the small temperature differences. The rate of heat transfer by convection is somewhat higher, but, owing to the usual low air velocities and small temperature differences, convection is not chiefly responsible for the cooling. The evaporation of a small amount of the water is responsible for most of the cooling.

When the temperature of a water surface is lower than the dry-bulb, but *higher than the wet-bulb temperature* of the surrounding air, heat will be transferred to the water surface from the air, principally by convection, but at a slower rate than that at which the water loses heat simultaneously by evaporation; hence the water becomes cooler in this case also.

In either case, the limiting temperature to which the water surface may be cooled by a flow of the atmospheric air over this surface is the wet-bulb temperature of the air. One might think that the water could be cooled to the dew point of the surrounding air as a limit, since evaporation continues until the water surface reaches this temperature; however, the rate of heat transfer from the air to the water exceeds the rate of heat loss from the water surface by evaporation as soon as the temperature of the water surface drops below the wet-bulb temperature. In fact, the wet-bulb thermometer of a sling psychrometer is the ideal water-cooling device.

(b) The **effectiveness** of the cooling system might be measured by the approach of the water temperature to the wet-bulb temperature of the air. In the A.S.M.E. Test Codes, the **cooling efficiency**, so called, is given as

$$e_c = \frac{t_1 - t_2}{t_1 - t'}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (671)$$

where t_1 = the temperature of the water before cooling, in deg. fahr.

t_2 = the temperature of the water after cooling, in deg. fahr.

t' = the wet-bulb temperature of the air, in deg. fahr.

The proposal has been made¹ to measure the effectiveness of the cooling in a different manner by the use of "cooling head." **Cooling head** is a measure of the available cooling effect for any water temperature and air condition. From a study of Eq. (661), the Ferrel psychrometric formula, it is seen that, for normal barometric pressure and the usual wet-bulb temperatures, the right-hand side of the equation reduces to about 0.011. This means that, in the cooling of the wet-bulb thermometer of a sling psychrometer, one degree of temperature difference between the air and the water on the bulb causes the rate of heat transfer by convection to balance the rate of heat utilization by evaporation due to a vapor pressure difference of substantially 0.011 in. Hg. Then with a water temperature of t_1 , an air dry-bulb temperature of t , and a wet-bulb of t' , the *evaporative* "cooling head" is the difference between the saturation pressure at the water temperature and the actual partial pressure of the water vapor in the air, or is $(p_1 - p)$; and the equivalent of the "cooling head" due to *convection*, in inches of mercury, is $0.011(t_1 - t)$. Thus, the *total* "cooling head" is $p_1 - p + 0.011(t_1 - t)$. But, for the wet-bulb equilibrium, $(p' - p)$ closely equals $0.011(t - t')$. The **total cooling head** at any stage of a

¹ "Water Cooling System Efficiency," by V. J. Azbe, Mech. Eng., Vol. 46, 1924, p. 799.

process in which water is being cooled by contact with air is, therefore,

$$c_1 = p_1 - p' + 0.011(t_1 - t'), \quad . \quad . \quad . \quad (672)$$

in which c_1 = the "cooling head," in in. Hg.

p_1 = the saturation pressure at the water temperature, in in. Hg.

p' = the saturation pressure at the wet-bulb temperature, in in. Hg.

t_1 = the temperature of the water, in deg. fahr.

t' = the wet-bulb temperature of the air, in deg. fahr.

Another measure of the effectiveness of water cooling, using "cooling heads" instead of temperatures, is

$$e'_c = \frac{c_1 - c_2}{c_1}, \quad . \quad . \quad . \quad . \quad . \quad (673)$$

in which c_1 = the "cooling head" corresponding to the temperature, t_1 , of the water before cooling.

c_2 = the "cooling head" corresponding to the temperature, t_2 , of the water after cooling.

(c) As an example to show the application of Eqs. (671) and (673), suppose that water is being cooled from 100 deg. fahr. to 80 deg. fahr with atmospheric air having a wet-bulb temperature of 60 deg. fahr.

From Eq. (671), the "cooling efficiency" is

$$e_c = \frac{100 - 80}{100 - 60} = 0.50.$$

From Eq. (673), the effectiveness of the cooling is

$$e'_c = \frac{1.931 - 0.521 + 0.011(100 - 60) - 1.031 + 0.521 - 0.011(80 - 60)}{1.931 - 0.521 + 0.011(100 - 60)} = 0.605.$$

586. Cooling Ponds.—When the amount of water to be cooled is small, and land space is cheap and available near by for constructing a shallow pond, or when there is a natural pond at hand, the water may be economically cooled by circulating it through such a body of water provided that an ample surface is exposed to the atmosphere. The pond should receive the warm water at the surface at one end and deliver the cooled water from near the bottom at the other end. The rate of cooling depends primarily upon the surface area of the pond, the difference in temperature between the water and air, the velocity and humidity of the air, and the length of air path per unit area of pond surface. The depth of the pond is usually from 2 to 4 ft., but the

volume of the pond should be sufficient to give storage capacity to meet variations in the load. The weight of make-up water, necessary to replace that evaporated during cooling, is usually not more than 2 to 5 per cent of the weight cooled.

Although experimental determinations of the performances of ponds are not in exact agreement, W. H. Carrier ² gives the following equation for the rate of evaporation of water into air flowing parallel to the water surface under standard barometric pressure:

$$w_e = 0.093 \left(1 + \frac{v}{230} \right) (p_s - p), \quad . \quad . \quad . \quad . \quad (674)$$

where w_e = the weight of water evaporated, in lb. per hr. per sq. ft. of pond surface.

v = the velocity of the air, in ft. per min.

p_s = the saturation pressure of water vapor at the water temperature, in in. Hg.

p = the partial pressure of the water vapor in the air, in in. Hg.

Then, the quantity of heat removed from the water due to this evaporation, in B.t.u. per hr. per sq. ft. of pond surface, would be

$$Q_e = 0.093r \left(1 + \frac{v}{230} \right) (p_s - p), \quad . \quad . \quad . \quad . \quad (675)$$

where r = the latent heat of vaporization at the temperature of the water, in B.t.u. per lb.

Since the temperature and the partial pressure of the water vapor in the air change as the air sweeps the pond surface, and since the temperature of the water is not uniform throughout the pond, these formulas give only approximate results. A safe empirical rule for the rate of heat removal from the pond water, in B.t.u. per hr. per sq. ft. of pond surface, is

$$Q_e = 150(p_s - p). \quad . \quad . \quad . \quad . \quad (676)$$

Example.—Find the area of pond surface necessary to cool 20,000 lb. of water per hr. from 140 deg. fahr. to 100 deg. fahr., with air having a dry-bulb temperature of 80 deg. fahr. and a wet-bulb temperature of 70 deg. fahr. with normal barometer, and find the weight of water lost by evaporation.

Using the mean temperature of the water and Eq. (676), the rate of heat removal is

$$Q_e = 150(3.44 - 0.62) = 423 \text{ B.t.u. per hr. per sq. ft.}$$

The area required is $\frac{20,000 \times 40}{423}$ or approximately 1900 sq. ft.

² Marks Handbook, 3rd edition, p. 1664.

Using an air velocity of 400 ft. per min., the mean temperature of the water, and Eq. (674), the weight of water evaporated is

$$w_e = 0.093 \left(1 + \frac{400}{230} \right) (3.44 - 0.62) = 0.72 \text{ lb. per hr. per sq. ft.}$$

The total weight evaporated per hour is about 1900×0.72 or 1370 lb.

587. Spray Ponds.—Spray ponds were developed in order to reduce the surface area of cooling ponds necessary for a given duty. In the spray pond system of water cooling, the water is forced under pressure through nozzles which distribute it through the air in a finely divided form before it is collected in the pond. This spraying of the water brings the minute particles into intimate contact with the air and greatly increases the surface exposed per unit weight of water cooled. The evaporation and consequent cooling proceed very rapidly, and for a given duty the spray pond requires only from 1 to 10 per cent as much surface as the simple cooling pond. This is an important advantage when the land available for the pond is expensive or the space restricted.

The **nozzles** or **spray heads** used are commonly of special nonlogging types, operating under gage pressures of from 3 to 15 lb. per sq. in. (7 lb. per sq. in. is common), and are generally spaced at 8- to 15-ft. intervals

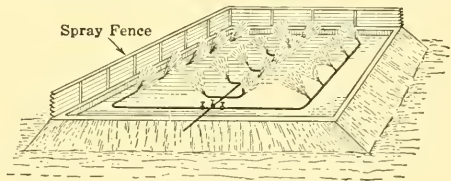


FIG. 970.—Spray Pond.

on rows of pipe from 15 to 20 ft. apart. The arrangement of the spray heads in a pond is shown in Fig. 970. Since an intimate contact of the water with the air is necessary in order to make the spraying effective, the arrangement of the nozzles should provide adequate air lanes through the spray. There must be ample pond surface beyond the spray to catch the drift in high winds, or, if the space is quite limited (as when the system is located on the roof of a building), fences with suitable air openings (**louvres**) must be provided to prevent excessive loss of water. As much as 6 per cent of the water cooled may be lost by evaporation and drift, even with these provisions. When a spray pond is used for cooling condensing water from a steam power plant, the energy consumed in pumping the water is from 1 to 2 per cent of the station output.

Additional cooling can be effected by **respraying** the pond water, either (1) by a mixed spraying system, in which pond water is added to the hot water in definite proportions before spraying, or (2) by independent respraying (double spraying) with the use of a separate

distributing system. Either method requires the use of more nozzles, larger or longer pipes, greater pond area, and greater expenditure of energy in pumping.

No general rules will be given for the selection or design of spray pond systems, because the great diversity of operating conditions, cost factors, local atmospheric conditions, and the like, makes each problem substantially an individual one.

588. Cooling Towers.—(a) A cooling tower consists of a wind-braced housing or *shell*, of wood, concrete, brick, or metal, enclosing a network of obstructions, or filling. The water to be cooled is pumped into a distributing system at the top of the tower, from which it is delivered in thin sheets or sprays to the filling; the filling is so arranged that the water is constantly spread out to expose new surfaces to contact with the air that flows through the tower. The cooled water is collected at the bottom of the tower in a reservoir or sump.

The air passing through the tower becomes partly or completely saturated by the evaporation of a portion of the water, and this evaporation is chiefly responsible for the cooling of the water. With an unlimited cooling surface and air supply, the water can be cooled to the wet-bulb temperature of the air as a limit. Actually, depending on the conditions of operation, the air is usually cooled to within 4 to 40 deg. of the wet-bulb temperature.

The cooling tower is often preferred to the cooling or spray ponds because it affords a greater cooling effect per unit of ground area occupied, can give a greater cooling range because of the longer time of contact of the air with finely divided water, and is more convenient for inspection and repairs. The changes in methods of cooling the condensing water used at the Hawthorne Works of the Western Electric Company³ at various times afford an illustration of what can be done when the cooling capacity has to be increased. The load on the original power plant was 2000 kw., and the condensing water was cooled in a simple cooling pond having a surface area of 70,000 sq. ft. The load gradually increased and spray nozzles were added over the pond until, when the load became 15,000 kw., a limit to the number of spray nozzles that could be placed over this reservoir was reached. At the present time, with a power plant load of 34,500 kw., the condensing water is cooled in two concrete cooling towers, as shown in Fig. 976 (page 1014), installed directly on the area occupied by the reservoir. In this installation, with the same ground area occupied by the three different water-cooling systems, the cooling towers carry over fifteen times the load of the simple cooling pond, and more than twice the load of the spray pond.

³ See *Power*, June 21, 1932, p. 907.

(b) The **filling** used in a cooling tower may consist of boards or lath, hollow tiles, metal sheets, or burlap. Cypress, redwood, and other woods that do not deteriorate rapidly in water are most commonly used. The water-distributing system and one arrangement of filling made up of triangular cypress lath are shown in Fig. 971. The evaporating surface of filling exposed per unit volume of the tower depends upon the type and arrangement of the filling, but usually is from 5 to 20 sq. ft. per cu. ft. of space filled, and the area free for air flow is from 65 to 85 per cent of the total cross-sectional area occupied by the filling. The velocity of air flow through the tower depends upon the type of tower and the operating conditions, but is usually from 100 to 700 ft. per min.

(c) The **various types of cooling towers** differ principally as to *arrangement of their housings* and as to the *draft-producing methods* used.

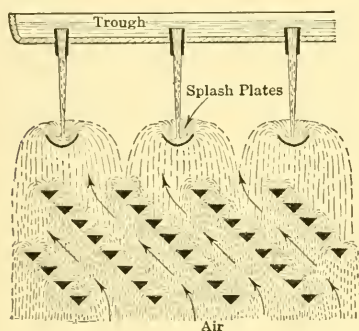


FIG. 971.—One Arrangement of Filling and Distributor. Filling of Triangular Cypress Lath.

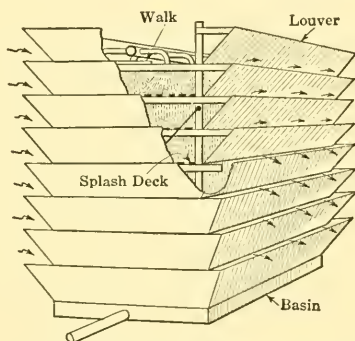


FIG. 972.—Atmospheric Cooling Tower.

They are generally classified as (1) atmospheric towers, (2) chimney-draft towers, (3) mechanical-draft towers, and (4) combined mechanical- and natural-draft towers.

(1) The **atmospheric cooling tower** (or open natural-draft tower), one arrangement of which is shown in Fig. 972, is the simplest form of tower. Air is admitted through louvred sides that prevent the water from being blown out, and flows through the tower, principally in a transverse direction. The rate of air circulation depends upon the wind velocity. Towers of this type are often designed to cool about 1.5 gallons of water per minute per square foot of active horizontal area with a wind velocity of 5 miles per hour. For effective cooling they are limited in width to but a few feet and, therefore, will not carry large cooling loads unless long. To prevent ice formation in winter, a sec-

ondary distributing system supplies water to only the lower portion of the filling.

(2) The **chimney-draft tower** (or closed natural-draft tower) has air inlets at its base, above which the sides are closed and extend, chimney-like, a distance of from 60 to 90 ft. above the filling, as shown in Fig. 973. The circulation of the air is independent of wind conditions and the flow is upward, counter to the falling water. For the same capacity, a chimney tower requires about one-half the ground area of an atmospheric tower. The draft is caused by the difference between the density of the column of air of high humidity and temperature within the tower and the density of the column of cooler and drier atmospheric air outside; and because of the better draft, the towers are particularly well suited for use with water of high initial temperature. The air velocity through the stack is commonly from 100 to 200 ft. per min., and the loss of water due to evaporation and drift

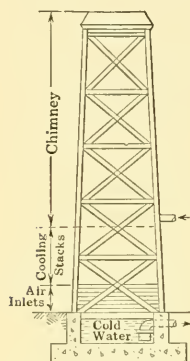


FIG. 973.—Chimney-Draft Tower.

is usually less than 3 per cent of the weight cooled. The stack portion is relatively expensive when the capacity of the tower is small, but is a minor item in the expense of a large tower.

A large chimney-draft tower is shown in Fig. 974. This tower, with a shell of reinforced concrete, cools five million gallons of screened sewage effluent per hour. The effluent is used to condense the steam exhausted from large turbines.⁴

(3) The **mechanical-draft cooling tower**, one form of which is shown in Fig. 975, resembles the chimney tower, except that it has a housing which is not so tall, has a more compact filling, and is equipped with one or more **forced-draft** or **induced-draft** fans, usually the former, to give positive circulation and control of the air supply. To prevent excessive loss of water and the nuisance caused by drift, the housing is frequently extended above the filling a distance of from 20 to 30 ft.; baffles may also be installed at the top of the tower structure to minimize the amount of water carried out by the air. For a given ground area,

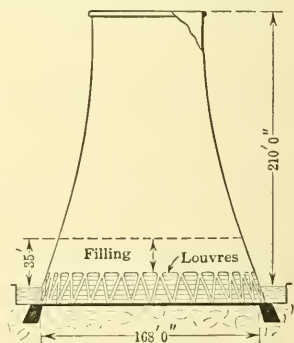


FIG. 974.—Large Chimney-Draft Tower

⁴ Engineering, Dec. 13, 1929, p. 759 and insert.

the mechanical-draft tower may have from two to five times the cooling capacity of a chimney tower. The velocity of air flow through the tower is usually between 300 and 700 ft. per min., and the loss of water by evaporation and drift should be under 5 per cent of the weight of water cooled.

Two large forced-draft cooling towers serving a 34,500-kw. power plant ⁵ referred to in (a) are shown in Fig. 976. Each of these towers is about 182 ft. long by 38 ft. wide by 50 ft. high, and is divided into 8 cells. As many cells as are necessary to carry the cooling load are then operated. Each cell is designed to cool 3800 gallons of water per minute from 106.5 deg. fahr., to 89.5 deg. fahr., and is equipped with four 8-ft.-diameter, two-blade propeller-type fans, each driven by a 15-h.p. motor. With this arrangement, each cell can cool the condensing water for a power plant load of about 2000 kw.

(4) The **combined mechanical- and natural-draft tower** permits the use of natural draft, with the fans shut off, when atmospheric conditions are favorable or when the cooling load is light; the mechanical draft is always available when needed.

589. The Energy Balance for a Cooling Tower.—

Using the methods explained in Sect. 582, an energy balance may be prepared for a cooling tower, when accurate and complete test data are available. The following notation will be used:

- w_1 = the weight of water entering the tower, in lb. per min.
- w_a = the weight of dry air passing through the tower, in lb. per min.
- t_1 = the temperature of the water entering the tower, in deg. fahr.
- t_2 = the temperature of the water leaving the tower, in deg. fahr.
- h_1 = the heat content of the entering water, in B.t.u. per lb.
- h_2 = the heat content of the exit water, in B.t.u. per lb.
- t_a = the dry-bulb temperature of the air entering the tower, in deg. fahr.
- t_b = the dry-bulb temperature of the air leaving the tower, in deg. fahr.
- w'_a = the weight of water vapor mixed with each pound of entering *dry* air, in lb.

⁵ See footnote 3.

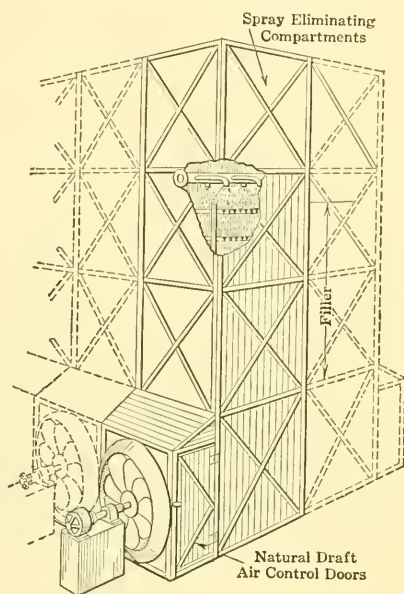


FIG. 975.—Forced-Draft Cooling Tower.

w'_b = the weight of water vapor mixed with each pound of exit *dry* air, in lb.

h'_a = the heat content of the air-water vapor mixture entering the tower, in B.t.u. per lb. of *dry* air.

h'_b = the heat content of the air-water vapor mixture leaving the tower, in B.t.u. per lb. of *dry* air.

Then, if external heat transfer, the loss of water by entrainment of moisture in the exit air, and differences in elevation and velocity

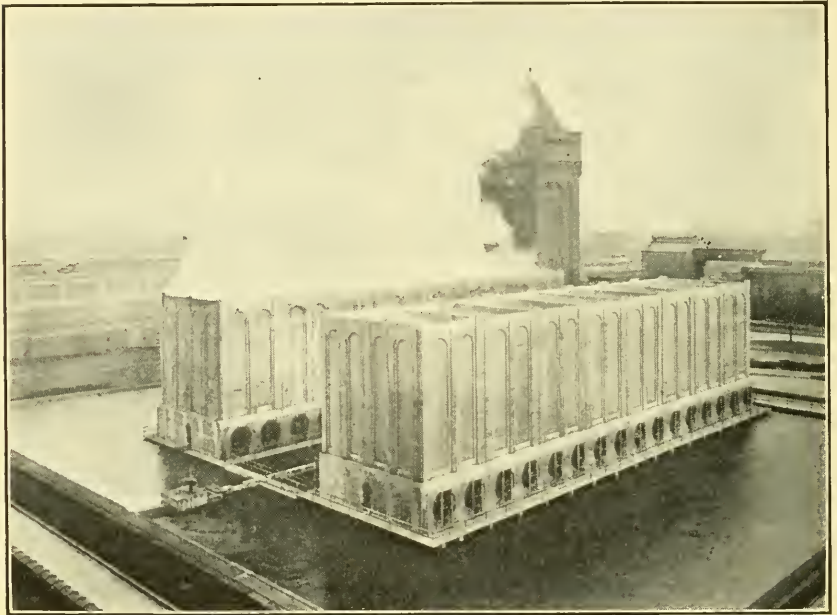


FIG. 976.—Large Forced-Draft Cooling Towers.

of the fluids at entrance and exit be ignored, the following equation expresses the energy balance:

$$w_1 h_1 - [w_1 - w_a(w'_b - w'_a)]h_2 + w_a(h'_a - h'_b) = 0. \quad (677a)$$

From which

$$w_a = \frac{w_1(h_1 - h_2)}{h'_b - h'_a - (w'_b - w'_a)h_2}. \quad (677b)$$

The measurements of states a and b should be taken between the respective fans and the tower, if fans are used; but if the small amount of energy required to operate the fans be neglected, a can be taken at

the entrance to a forced-draft fan and b at the discharge of a fan of the induced type.

As an **example** to show the use of Eq. (677*b*), a typical numerical case will be solved. A cooling tower must cool 3800 g.p.m. from 106 deg. fahr to 89 deg. fahr. when the atmospheric dry-bulb temperature is 80 deg. fahr, the wet-bulb temperatures 70 deg. fahr., and the barometric pressure is 28 in. Hg. If the air leaves the tower at a temperature of 95 deg. fahr with R.H. = 1.0, find the weight and volume of *dry* air that must pass through the tower per minute, and find the amount of water evaporated in percentage of the weight of water entering the tower.

From Eq. (662), $w'_a = 0.0143$ and $w'_b = 0.0391$. Then from Eq. (677*b*), the weight of *dry* air passing through the tower is

$$w_a = \frac{31,500 \times 17}{75.9 - 44.6 - 0.0248 \times 57} = 17,900 \text{ lb. per min.}$$

The volume of dry air passing through the tower per minute is, then,

$$(17,900 \times 53.3 \times 540) \div (27.37 \times 0.491 \times 144)$$

or 267,000 cu. ft. per min. The weight of water evaporated, in percentage of the weight of water entering the tower, is $\frac{17,900 \times 0.0248}{31,500} \times 100$ or 1.4 per cent.

590. The Design of Cooling Towers.—(a) Cooling towers have been designed largely on empirical information and in accordance with previous successful designs. Sufficient filling surface must be provided to carry the required cooling load with the worst atmospheric conditions, and the filling must be so arranged that there is not an excessive resistance to air flow through the tower. The velocity of the air through the air passages in the tower is of primary importance in determining the rate at which heat can be removed from the water.

A cooling tower is usually guaranteed to cool a definite weight of water per unit time through a definite temperature range with definite dry-bulb and wet-bulb temperatures of the atmosphere. With atmospheric towers, the wind velocity, and with mechanical-draft towers, the fan speed, are also specified. A guarantee may also be made concerning the amount of make-up water required and the power consumed by the auxiliary pumps and fans. In testing the tower, the atmospheric condition is beyond control and is usually not the same as the condition specified in the guarantee. The question then will arise as to whether the equipment is meeting the guarantee; hence it is desirable to have some previous agreement as to the effect of deviation of the atmospheric condition from the guarantee upon the performance of the tower.

(b) In order to give some basis for design and to determine the effect of the different variables upon the performance of a cooling tower, a principle similar to that used in heat transmission is suggested. This

principle has previously been touched upon by Coffey and Horne ⁶; and, in different form, by Robinson.⁷ If the **performance factor** of a cooling tower be defined as the quantity of heat removed from the water per unit time per unit effective area of filling per unit of mean “cooling head,” then the following equation, similar to Eq. (432), may be written:

$$q = V_F \left(\frac{A_F}{V_F} \right) F c_m \quad . \quad . \quad . \quad . \quad . \quad . \quad (678)$$

where q = the quantity of heat removed from the water, in B.t.u. per min.
 V_F = the volume of the filling, in cu. ft.
 $\left(\frac{A_F}{V_F} \right)$ = the area of filling surface in contact with water per cu. ft. of volume, in sq. ft.
 c_m = the mean “cooling head,” in in. Hg.
 F = the performance factor, in B.t.u. per min. per sq. ft. of effective filling surface, per in. Hg of mean “cooling head.”

Then the performance factor, F , is analogous to the overall coefficient of heat transmission, U , in heat-transfer work. The mean “cooling head” is taken as the arithmetic mean of the “cooling heads” at water entrance to and water exit from the tower. The arithmetic mean is probably satisfactory, since the usual difference between the “cooling heads” at entrance and exit is not large. In accordance with the definition of Sect. 585(b), the “cooling head” at water entrance, c_1 , and at water exit, c_2 , for a counter-current tower may be found as follows:

$$c_1 = (p_{t_1} - p_{t'_b}) + 0.011(t_1 - t'_b), \quad . \quad . \quad . \quad . \quad (679)$$

$$c_2 = (p_{t_2} - p_{t'_a}) + 0.011(t_2 - t'_a), \quad . \quad . \quad . \quad . \quad (680)$$

in which p is the saturation pressure of water vapor at the temperature indicated by the subscript, in in. Hg. The arithmetic mean “cooling head” is, then,

$$c_m = \frac{c_1 + c_2}{2}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (681)$$

The performance factor, F , has been calculated for several tests on a slat-filled cooling tower reported by Robinson.⁸ Data were not given for the cross-sectional area free for air flow through the filling nor for

⁶ A.S.R.E. Journal, Vol. 1, No. 1, p. 78.
⁷ Trans. A.S.M.E., Vol. 44, 1922, p. 669.
⁸ Ibid.

the effective filling surface per unit volume of filling. The dimensions of the tower were reported as 10 ft. by 10 ft. by 22 ft., so it was assumed that 80 sq. ft. of cross-sectional area was free for air flow and also that 8 sq. ft. of filling was in contact with the water for each cubic foot of filling volume. The resulting performance factors, F , and air velocities, v , through the tower are not absolutely correct, but the calculations shown in Table LXXXIII indicate what might be done with complete test data. The dependence of the performance factor on the air velocity through the tower is shown in Fig. 977. The performance factor for this tower increases with the 0.86 power of air velocity, regardless of whether forced or natural draft is used. The ratio of performance factor to the 0.86 power of air velocity, $F/v^{0.86}$, is shown in the last column of the table, and the ratio for this tower is substantially constant regardless of air conditions, cooling range, or draft.

If the tests of cooling towers were completely reported, the performance factor of different types of towers could be determined, and this factor could then be used in cooling-tower design in the same way as the overall coefficient of heat transmission, U , is used in the design of heat-exchange apparatus. Also, the influence of primary variables, such as the air velocity, upon this factor could be determined. The design of cooling towers might then proceed on a more rational basis, the performance under any set of operating conditions could be more closely predicted, and the question of whether a tower is meeting guaranteed performance when the atmospheric conditions are different from those of the guarantee could be answered.

591. The Selection of a Water-Cooling System.—The selection of the type and size of a cooling system for a given duty depends not only upon performance, but also upon the first cost and operating costs. The cooling system can seldom be considered independently; instead, its relation to the apparatus it serves must be studied. For example, an expensive cooling system may be installed that will cool condensing

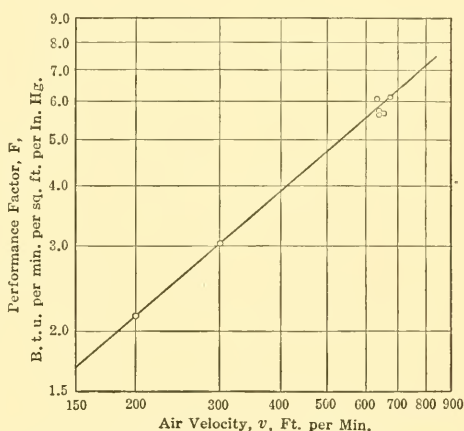


FIG. 977.—The Relation between Performance Factor and Air Velocity for One Tower.

TABLE LXXXIII

PERFORMANCE OF A 10 FT. BY 10 FT. BY 22 FT. SLAT-FILLED COOLING TOWER

Test No.	Draft	t_1 Deg. Fahr.	t_2 Deg. Fahr.	t_a Deg. Fahr.	t'_a Deg. Fahr.	$t_b=t'_b$ Deg. Fahr.	q B.t.u. per Min.	c_1 In. Hg	c_2 In. Hg	c_m In. Hg	Perform- ance Factor F	Air Veloc- ity, v , Ft. per Min.	$F \frac{p^{0.86}}{p^{0.86}}$
1	Forced	105.0	84.7	71	57	90	110,500	0.99	1.04	1.02	6.18	675	0.023
2	Forced	107.8	87.5	72	63	93	108,000	1.02	1.00	1.01	6.07	633	0.024
3	Forced	112.0	88.5	64	56	96	124,500	1.21	1.26	1.24	5.73	638	0.022
4	Forced	108.5	87.0	69	57	92	115,000	1.15	1.15	1.15	5.69	650	0.022
5	Forced	109.9	90.5	83	69	95	103,400	1.09	0.97	1.03	5.70	638	0.022
6	Natural	116.0	98.0	43	40	101	94,800	1.25	2.21	1.73	3.11	300	0.023
7	Natural	135.0	115.8	60	55	118	102,000	2.09	3.29	2.69	2.15	200	0.023

water to within a few degrees of the wet-bulb temperature of the air. Whether or not this is economically desirable will depend upon the saving in the cost of the fuel burned to generate the steam for the prime mover that results from the reduction in condenser pressure, as contrasted with the high fixed costs and high operating costs of the water-cooling equipment.

CHAPTER XLIX

THE STEAM POWER PLANT ENSEMBLE; LOCATION, ARRANGEMENT, ECONOMICS, COSTS

592. General Factors Affecting Plant Location and Design.—

(a) The location of a plant should be selected with respect to (1) nearness to the center of demand for the plant output (electric or mechanical power, or steam for industrial uses); (2) facilities for receiving fuel (preferably by at least two independent transportation systems, one being water which is often the cheapest, thus avoiding the possibility of simultaneous interruptions due to strikes or accidents); (3) space for the storage of a reserve supply of fuel; (4) adequacy of water supplies suitable for condenser cooling and for boiler feed (preferably more than one source for the latter); (5) cost of land and its preparation; (6) suitability of the ground for the foundations of the building and equipment; (7) disposal of ashes, as by filling neighboring swamps or gullies to make waste land useful; and (8) the possible opposition which may be raised by property owners in the neighborhood to the chimney discharges. In most cases the site selected is the result of a compromise dictated by local conditions and by the cost factors involved. The ideal location for a steam-electric power plant using coal as the fuel, is at the mouth of a mine and in the immediate vicinity of a large body of water which may be drawn upon to any required extent for condenser cooling and for make-up boiler feedwater, and also near the center of demand for the output; however, this combination of advantages is seldom met, and consequently the location depends upon an analysis of the factors just given.

(b) The structure of a plant may be either simple or complex, and the appearance may be modest, ornate, or even monumental, depending upon actual needs, upon requirements of harmony with neighboring structures, or upon the designer's fancy. Unless the building is very simple, its designing and construction usually require the services of architects and civil engineers. Under favorable climatic conditions the enclosing structure may be of the simplest and cheapest kind, and it may be dispensed with altogether in some cases; often small temporary plants, such as those used by contractors, are left uncovered, and even

some large steam-generating plants and some mercury-vapor plants have been built in the open.¹ Most plants have the steam-generating units and the prime movers located in separate enclosed rooms, but in several modern plants the wall between the rooms has been omitted.

The design of the plant structure depends to a large extent upon the size, shape, and location of the equipment which is to be enclosed; and carefully coordinated planning is necessary in order that adequate clearance may be provided between the parts of the structure and the equipment, and that the repairing of the apparatus may be facilitated.

Structural-steel shapes are commonly used for the framework of the building and also for supporting a considerable portion of the equipment. Even a moderate-sized plant requires a large amount of structural steel, as shown by Fig. 978; and in this figure it will be seen that such equipment as the coal bunker and the chimney are supported by the framework, while the steam-generating units (not visible) rest upon girders of their own. In most plants the building housing the original group of equipment is closed at one end by means of a temporary curtain wall which can be removed when the capacity of the plant is increased by the addition of

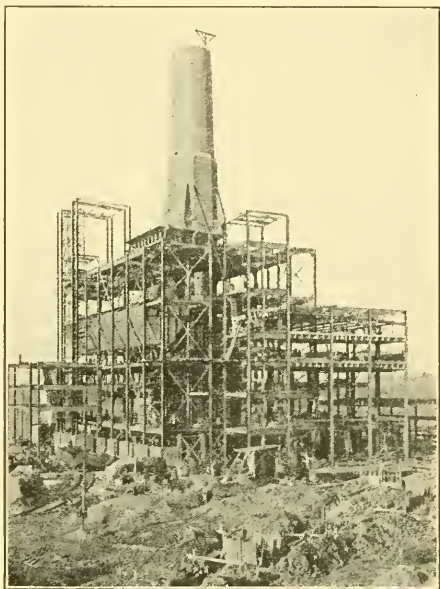


FIG. 978.—Power Plant at the Point Breeze Works of the Western Electric Co.

more apparatus and an extension of the structure. Usually it is better to allow for plant expansion in this way than to provide extra space in the original building, since the location of columns and walls in the first structure may eventually prove to be unsuitable for the new requirements resulting from changes in the art which cannot be foreseen.

Equipment foundations should not only be designed to withstand the dead weights imposed upon them but should also be massive enough to absorb vibrations effectively. Piles must be used when either

¹ See "Features of the Outdoor Mercury-Vapor-Steam Plant at Schenectady," in *Power Plant Eng.*, Aug., 1933, p. 354.

swamps or quicksand are encountered, and these columns are usually surmounted by a thick mat of reinforced concrete. If the plant is located on the shore of a river where it must withstand floods, the foundation and walls may be so constructed as to constitute, in effect, a massive "barge" which will not float; in this case the bottom of the "barge" is a reinforced concrete mat from 4 to 6 feet thick, which serves also as a foundation for the equipment.

(c) The **complexity of a plant** depends upon the efficiency of energy conversion which it is expected to show; and this, in turn, is usually related to economic requirements. The simplest steam power plant consists of an internally fired boiler which is fed by an injector and which is combined with a steam prime mover to form an integral unit; examples of such plants are to be found in small portable or semi-portable units, steam tractors, "locomobiles," and the like. Next in complexity are the moderate-sized non-condensing plants with hand-fired boilers. These simple plants far outnumber the larger and more complicated industrial plants and central stations which operate condensing and contain a variety of auxiliary equipment to assist in producing the output at the lowest feasible cost. Each piece of equipment added to raise the efficiency also increases the complexity and cost of the plant. In any case the equipment installed, and its arrangement within the plant, should be as simple as possible, consistent with the requirements to be met.

(d) The **arrangement of equipment** in a plant depends somewhat upon the extent and shape of the ground available; commonly, however, the plot is chosen so that the back-to-back arrangement of steam-generating units and prime movers, or a modification thereof, may be employed. In large plants each combination of a prime mover with its steam generators and the auxiliary apparatus is laid out as a unit, the plant as a whole consisting of duplicate units with such interconnections between them as are necessary. If the prime movers are turbines, their axes may be parallel or at right angles to the wall (if any) between the turbine and boiler rooms; if the prime movers are reciprocating engines, the longitudinal axes through the cylinders are almost invariably placed perpendicular to this wall, making a true back-to-back arrangement. The axes of turbines and of their condensers may be parallel or crosswise with respect to each other; and the space around each of the units must be sufficient to provide room for the cleaning and renewal of condenser tubes. Prime mover and condenser auxiliaries are usually placed in the basement immediately adjacent to the units they serve. The turbine or engine room is always clear above the main floor level and is provided with an overhead traveling crane of lift and

capacity suitable for handling the largest and heaviest parts of the equipment.

The steam generators can be placed with the firing aisles either parallel or perpendicular to the wall separating them from the prime movers. While formerly in some very large plants, many small steam-generating units were used and, where ground was expensive, were placed on two floors (as in the New York Edison Waterside Stations, Nos. 1 and 2), in modern plants these units are relatively large and few in number and are placed on the ground level, the tendency in multiple-unit plants being toward the use of but two steam generators, or even one, with each turbine. The steam-generator room, commonly called the "boiler room," contains, in addition to the steam generators themselves, the auxiliaries serving them, the flues, and the equipment for the indoor handling of coal and ashes; and sometimes the base of the stack is located in this room, or on its roof. The room and its openings should be so arranged, and the equipment within it so disposed, that the necessary space is provided for the replacement of tubes in boilers, economizers, and air heaters.

Both the turbine (or engine) room and the boiler room should have doorways large enough to admit the various pieces of equipment; and frequently railroad tracks are extended into the end of the turbine room so that cars can be unloaded by means of the crane. Provision must be made also for housing the electrical equipment, when that is necessary; and space for this purpose may be provided in the main structure, in a separate electrical building, or out of doors. (Some of these features mentioned in this and the preceding paragraphs are shown later in Figs. 979 to 985, illustrating typical layouts.)

(e) The **selection of plant equipment** must, in general, be made with respect to the thermal, operating, and cost factors involved. In the preceding chapters these factors have been discussed for each of the principal kinds of equipment; however, when equipment is to be arranged to form a complete plant, the choice of apparatus depends upon the way in which the various factors just mentioned are affected by the *entire combination*, or system, of which each piece of equipment is a part, when the plant is functioning at all of the normal rates of output. The main systems to be considered in planning the equipment are those concerned with (1) the coal and ash handling, (2) the draft, (3) the steam supply to the prime movers, (4) the condensing water, (5) the exhaust steam from the main units, and the resulting condensate, (6) the boiler feedwater supply, (7) the feedwater heating, and (8) the piping arrangements necessary to handle the various fluids employed. In addition to the systems just listed, consideration must

be given to (1) the type of auxiliary drive (electric, steam, or dual), (2) the handling of the steam supply to, and the exhaust steam from, the steam-driven auxiliaries, and (3) the control system. All of these

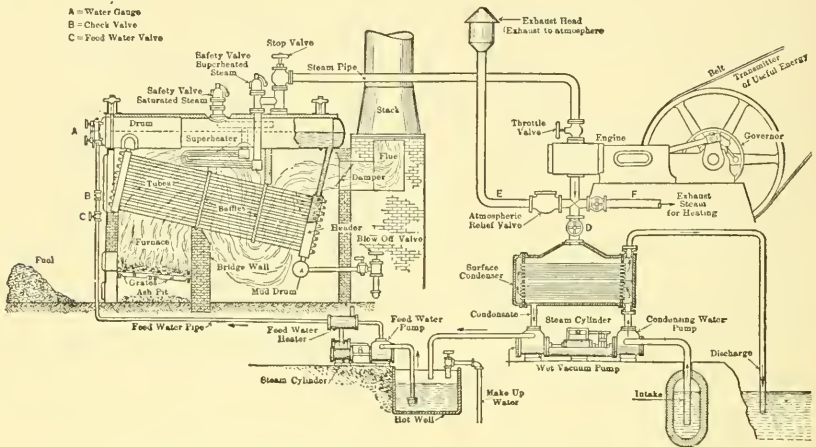


FIG. 979.—Elements of a Simple Steam Power Plant.

systems should be so designed that, when functioning together, the desired operating conditions will be obtained; furthermore, the combination employed should involve the least initial investment consistent with the greatest convenience and economy of operation.

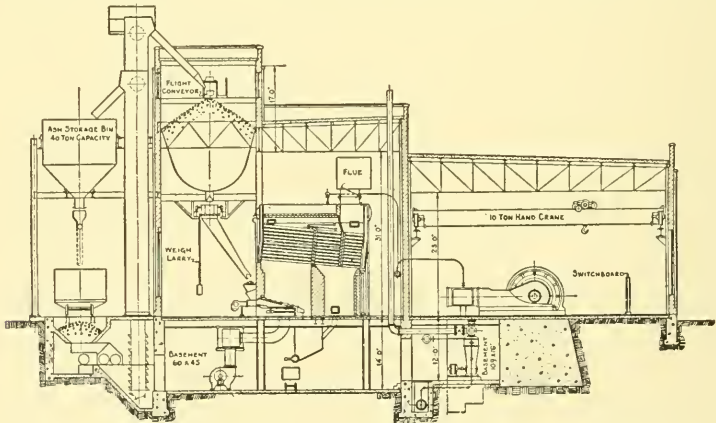


FIG. 980.—Cross-section of a 1500-Kw. Plant with Compound 4-Valve-Engines.

593. Some Typical Steam Power Plants.—(a) A few examples of power-plant arrangements are given in Figs. 979 to 985. The first figure shows the elements of a very simple condensing plant with

steam-generating unit, reciprocating engine, condenser and its auxiliary pumps, feed pump, and heater.

(b) The cross-section of a somewhat more elaborate **reciprocating-engine** plant is given in Fig. 980, which shows, in addition to the arrange-

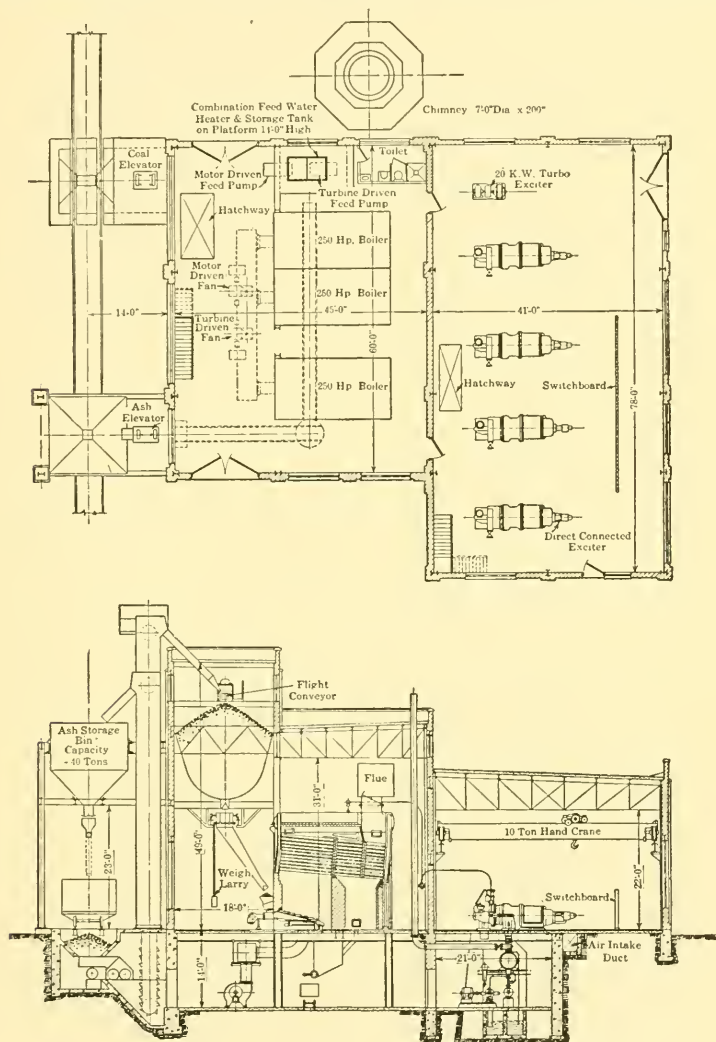


FIG. 981.—Layout of a 1500-Kw. Turbine Plant.

ment of the main power equipment and the building, the simple coal-and-ash and forced-draft systems. This plant has four 500-kw. engines and three steam-generating units. One engine and steam generator

are spares; hence, although 2000 kw. are installed, the normal capacity is 1500 kw. The plan of this plant is substantially the same as that shown in Fig. 981.

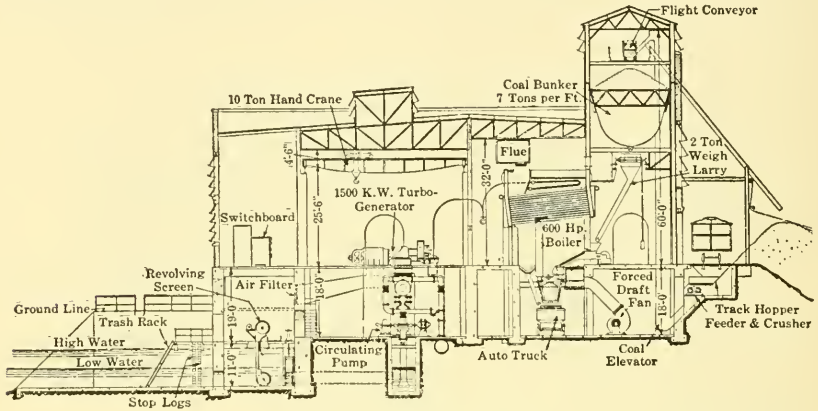


FIG. 982.—7500-Kw. Turbine Plant.

(c) The layout of a 1500-kw. turbine plant of the same total capacity as the engine plant described in (b), is shown in Fig. 981, the two

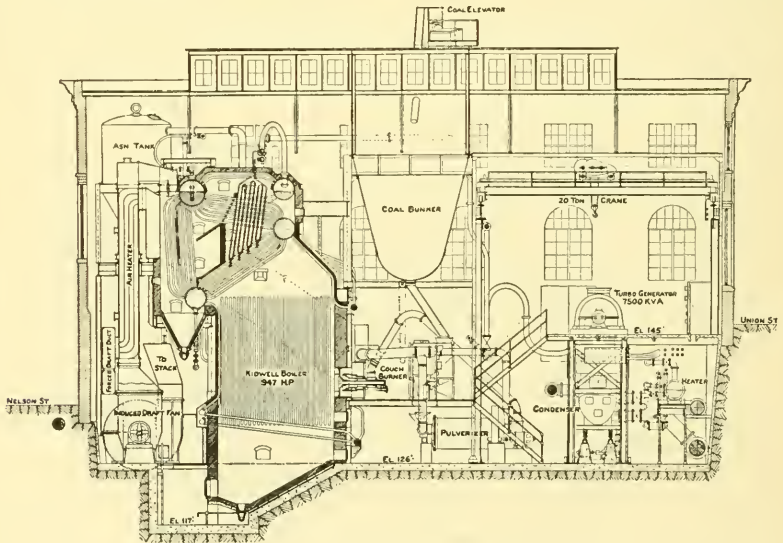


FIG. 983.—A Small Compact Layout.

plants being competitive designs. Neither plant employs the regenerative cycle, because of its small size. The feedwater is normally heated

by exhaust from the auxiliary drives, used in a feedwater heater located near the stack.

(d) The cross-section of a 7500-kw. turbine plant is given in Fig. 982,

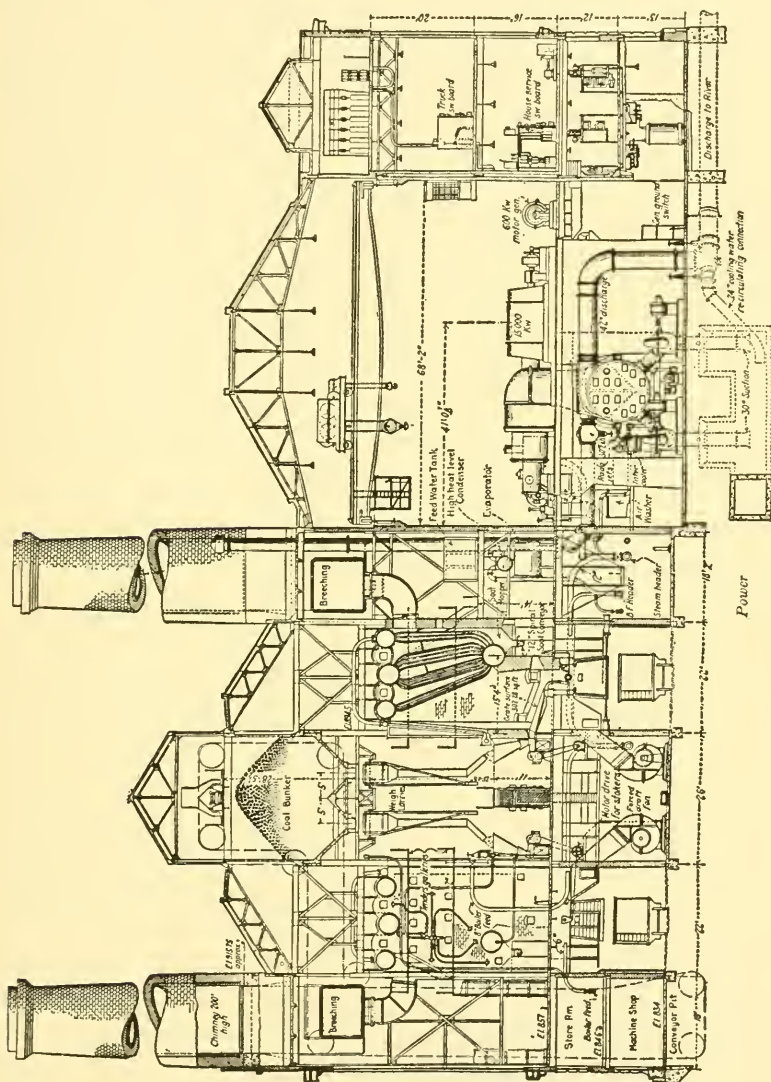


FIG. 984.—Moores Park Station, Lansing, Mich.

which shows in addition to the other features, the method of supplying the condensing water and the generator cooling air. The plant has four 2500-kw. turbines and four steam-generating units, and operates

on the Rankine cycle. The feedwater is heated in the same manner as in the preceding case.²

(e) A **very compact layout** for a 7500-kw. installation, added in the restricted space available in an old plant, is shown in Fig. 983. The steam generator is fired by means of powdered coal; forced and induced draft are used, and the air for combustion is preheated. The turbine has one extraction heater; and an evaporator and a deaerating heater, neither of which is shown, are also included in the feedwater heating system.³

(f) A **municipal central station**⁴ is shown in cross-section in Fig. 984. There are two rows of steam-generating units with a central firing aisle between, and two chimneys supported at roof level. Most of the important features are clearly shown and are labeled. The arrangement of plant is quite typical of that used with many of the older ones.

(g) The **Gilbert 1200-lb. station** (originally known as the Holland station) is shown in transverse section in Fig. 985. The small 1200-lb. high-pressure unit has a capacity of 12,000 kw. and normally furnishes steam at 400 lb. per sq. in. to the large 43,000-kw. turbine generator. The two steam-generating units are substantially the same as that shown in Fig. 539, p. 434. The plant has four bleeder heaters, one being a deaerating heater, and an evaporator condenser with a vent condenser, in the feedwater heating system, in addition to the economizer.⁵ This is a base load plant, operating in conjunction with other plants.

594. Combinations of Apparatus for Conserving Energy.—(a) **Systems of plant equipment** usually effect a degree of energy conservation somewhat in proportion to their complexity. It should be noted, however, that although the use of more efficient equipment, or the addition of heat-recovery apparatus, will always result in an improvement in plant performance, the inclusion of such equipment may not be economically sound, as will be shown later. Several examples of the limitations and advantages of various systems of plant equipment follow:

(1) The *simplest steam power plant*, consisting of a non-condensing

² "The Design and Maintenance of a 1500-Kw. Central Station Plant," Proc. N.E.L.A., 1924, p. 1000; and "Design and Maintenance of a 7500-Kw. Central Station Plant," Proc. N.E.L.A., 1924, p. 1413.

³ "Increase of Steam Power Facilities of the New Brunswick Power Co.," by J. D. Garey, presented to the Engineering Institute of Canada, Dec. 12, 1929.

⁴ "Municipal Power Plant," by O. E. Bulkeley, Power, June 19, 1923, p. 968; and "Moores Park Completed," Power Plant Engineering, Nov. 1, 1931, p. 1048.

⁵ "The Holland Station," by E. M. Gilbert, G. E. Review, Feb., 1929, p. 95; "Design of Holland Station," Power, May 28, 1929; "The Gilbert Station," by T. G. Thurston, National Engineer, March, 1931.

prime mover and a steam generator fed by an injector, operates on an "open system" which wastes the exhaust steam; and in this case the addition of an exhaust-steam feedwater heater will effect a fuel saving of, say, 10 to 14 per cent. However, only a portion of the exhaust steam is needed for feedwater heating, the rest being wasted; hence a good thermal performance cannot be obtained in a plant of this kind, although a considerable improvement results from the feedwater heating. It should be noted that the addition of the heater may require the substitution of a mechanical feed pump for the injector and that a thermal gain is obtained only with an increase in the cost and complexity of the plant.

(2) In a *simple condensing plant* a thermal gain will result from the

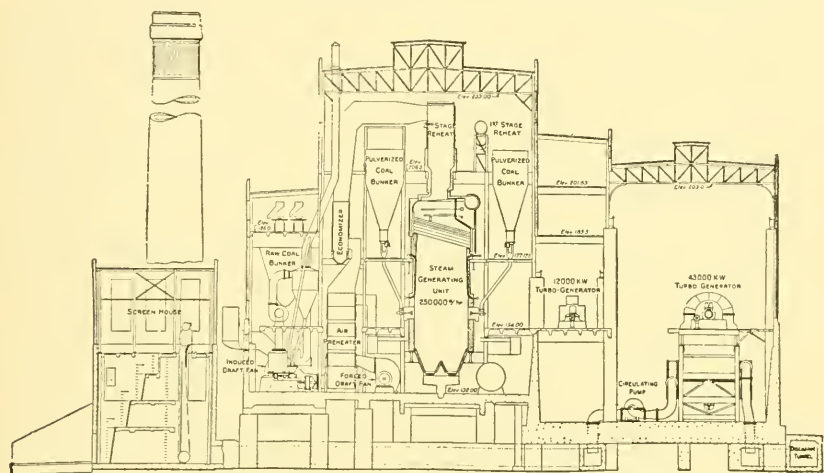


FIG. 985.—Gilbert 1200-Lb. Station, Holland, N. J.

reduction in the back pressure through the use of a condenser, and a further improvement in performance may be obtained by abstracting heat from the exhaust steam in a feedwater heater of the through type; however, the cost of the latter usually does not justify its installation. But in this type of plant a considerable saving may be effected by using steam-driven auxiliaries and employing their exhaust steam in a closed heater to raise the temperature of the condensate from the main condenser, this water being recovered and returned to the boiler through the heater.

(3) The use of a *combination of steam- and electrically driven auxiliaries* in an ordinary condensing plant is a method for obtaining a satisfactory energy balance with varying loads. This balance is effected

by shifting from steam to motor drive, and vice versa, as the load conditions change, the motors taking their current from the main generators.⁶

(4) A *house turbine-generator* may be employed to furnish current for electrically driven auxiliaries, the exhaust steam from this unit being discharged into a barometric or jet condenser in which the feedwater is used for condensing purposes. This system has frequently been installed for the purpose of effecting an improved energy balance, and it has the added advantage that the feedwater may be degasified without the addition of special equipment to accomplish this. The house turbine may be arranged to produce current for all of the auxiliaries or only a portion of them; and in the latter case some of the auxiliaries are steam-driven. In either case the common practice is to arrange the electrically driven auxiliaries so that current may be supplied to them from either the house turbine or the main units.⁷ With this last system the operator can load the house turbine in such a way that it will at all times furnish the proper amount of exhaust steam to produce the desired energy balance.⁸ If the quantity of steam exhausted by the house turbine is insufficient at certain loads to supply the demand for feedwater heating, a supplementary supply for this purpose may be obtained by bleeding steam from intermediate stages of the main turbines.⁹

The four hook-ups of equipment just described are such that the performance of the plants may be referred to that of the *Rankine cycle* as a criterion. In general, a simple hook-up is preferable in small or moderate-sized plants, and in large ones which are frequently operated at low percentages of their capacity or have long periods of interrupted service. However, a more complex arrangement than any of those just described may be profitably employed in plants such as the large base-load stations, which operate at high percentages of their capacity for long periods of time; hence such plants are arranged to function on either the regenerative or the regenerative-reheating cycle.

(b) The **regenerative cycle** is used to effect a greater conservation of energy than is possible with any of the systems described in (a);

⁶ For a discussion of a system of this kind, see "Auxiliary System and Heat Balance at the Delaware Station of the Philadelphia Electric Co.," Trans. A.S.M.E., v. 43, 1921, p. 475.

⁷ See "Heat Balance of the Connors Creek Plant of the Detroit Edison Co.," by C. H. Berry and F. E. Moreton, Trans. A.S.M.E., v. 43, 1921, p. 500; also "Heat Balance of Colfax Station," by C. W. E. Clarke, *ibid.*, p. 487.

⁸ See "Heat-Balance System for Hell Gate Station," by J. H. Lawrence and W. M. Keenan, Trans. A.S.M.E., v. 43, 1921, p. 495.

⁹ See Power, May 13, 1924, p. 750.

but its use is accompanied by a greater expense for the numerous heaters employed, together with the added piping, pumps, and traps. When the prime movers are small the cost of this added equipment is relatively large; and if extraction heating is employed in this case, it is not economical to bleed steam from more than one or two stages of the main turbines. When the prime movers are large the cost of the added equipment is proportionately less, and it then becomes economical to provide 3, 4, or even 5 stages of regenerative heating, the proper number depending upon the range of steam pressure and temperature in the main units, the character of the load, the presence of other heat-recovery apparatus in the system, and other operating conditions. *Reheating*, which is used in combination with the regenerative cycle when the initial steam pressure is high, is employed not so much for the purpose of increasing the conservation of energy in the plant as to prevent the presence of an excessive amount of moisture in the steam passing through the low-pressure stages of the main turbines. Evaporators and deaerators, when included in the plant, are placed in the system with the other feed heating apparatus and use steam bled from the main units or received from other sources; and in addition to these, the plant system may include various other heat exchangers such as air-ejector condensers, generator air-coolers, oil coolers, and blowdown heat exchangers.

595. Regenerative Feedwater-Heating Systems.—(a) The number, type, and arrangement of the heaters used in the regenerative system determine, to a large extent, the energy balance of the plant in which they are installed. Both closed and direct-contact heaters are used in these systems; most of the plants employing extraction heaters have been fitted with the closed type of apparatus, but a few plants of advanced design are equipped entirely with the direct-contact type, while several installations include both types in combination. The characteristics of the heaters themselves have been discussed in Chapters XLV and XLVI; and each type has certain advantages and limitations when used in a complex plant system with other equipment.

(b) **Closed heaters** have been extensively used in the past because when they are installed there is almost no likelihood of feedwater entering the turbine through the bleeder connections. Owing to the frictional resistance of the tubes, however, the work of the pumps is greater with this type of heater; and if the size of the heater is increased in order that the terminal temperature difference between the feedwater and the entering steam may be reduced, consideration must be given to the greater cost of this larger heater as well as to the increased expense of pumping against the higher resistance. A representative modern

plant completely equipped with closed heaters¹⁰ is shown in Fig. 986; and it will be seen that steam is bled from four intermediate stages of the main turbine for use in four heaters through which the feedwater flows in series. In this system the proper drainage of the heaters requires the use of several pumps and traps, the operation of which will be discussed later. Since these heater drain pumps discharge into the main feed line, care must be taken to connect them so that their delivery pressures will be above the flash point for the temperature of the water being pumped, otherwise trouble will be experienced with them, due to their becoming steam bound. The flow diagram shown in Fig. 986 includes an evaporator and its accompanying condenser which are so

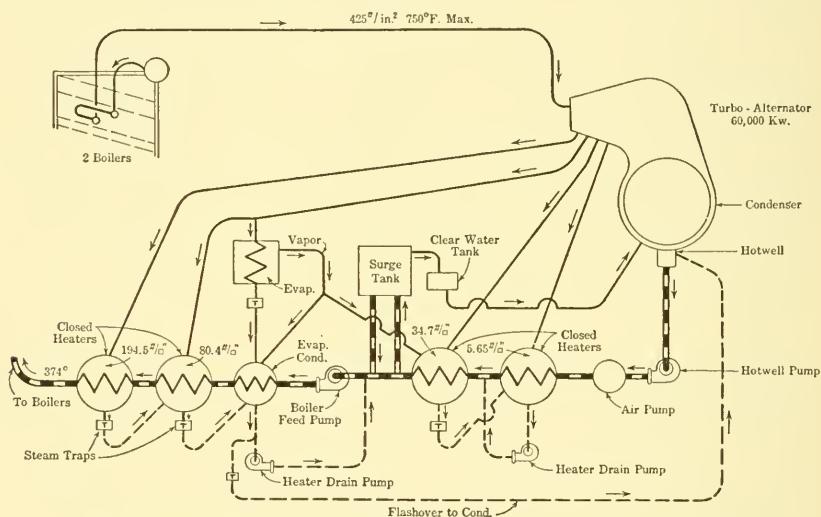


FIG. 986.—Flow Diagram of the Atkinson Plant, Georgia Power Co.

arranged that the vapor produced in the evaporator is condensed by giving up heat to the feedwater, while both the condensed vapor and the condensate from the evaporator coils finally enter the main feed line. The boiler feed pump discharges through two heaters which must therefore be designed for high pressures.

(c) A complete installation of **direct-contact heaters** is shown in Fig. 987; and this system also includes an evaporator with its condenser, as well as several other heat exchangers. In this case each heater is fitted with a vent condenser, the condensate from which drains into the heater discharge; and each heater is also served by

¹⁰ For a complete description see the Southern Power Journal, Oct., 1930; also Power, Oct. 14, 1930, p. 604.

a condensate pump, the boiler feed pump being placed after the last heater. There is copious provision for water storage in the various

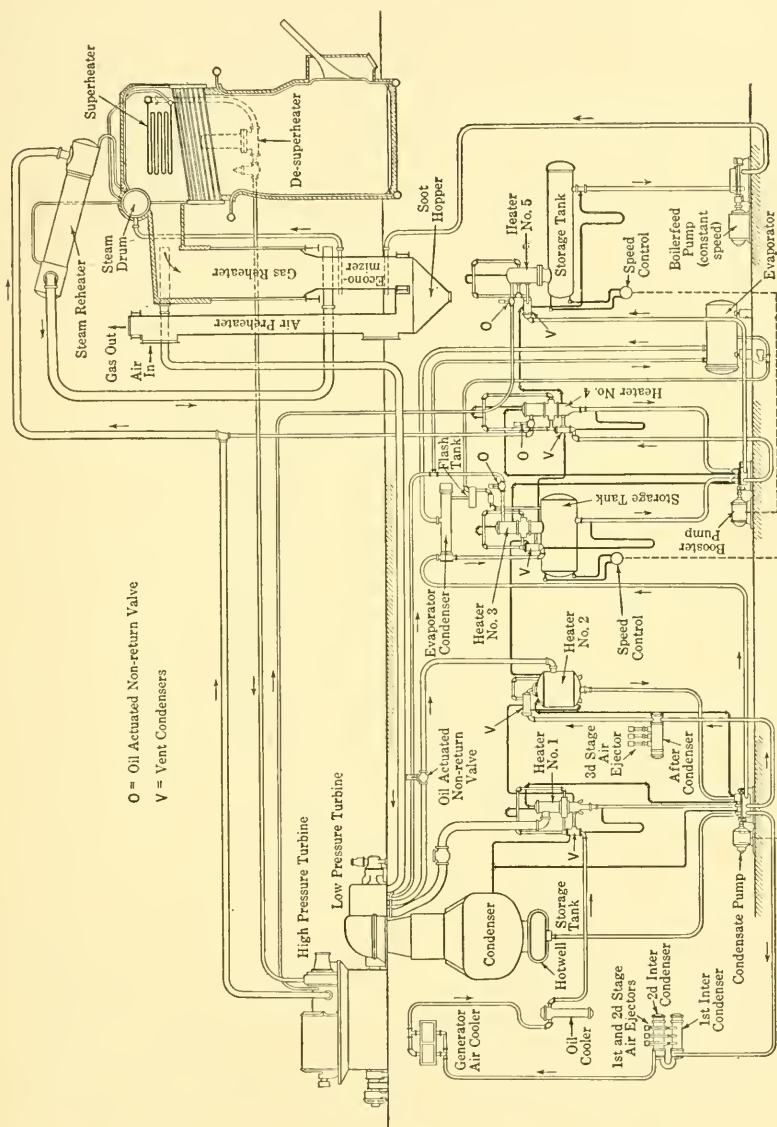


Fig. 987.—Flow Diagram of the South Amboy Station, Jersey Central Power and Light Co.

heaters and in the hotwell to overcome one of the difficult problems associated with a direct-contact-heater system, namely, water regulation. The plant illustrated in Fig. 987 also has an interesting combina-

tion of live-steam and gas reheating (similar to that described on p. 435) to assist in producing the desired energy balance.¹¹

(d) A combination of **direct-contact and closed heaters** is shown in Fig. 988; and in this plant¹² it will be seen that there are four stages of extraction heating, the direct-contact heater forming the third stage. In this system the boiler feed pump is placed after the third stage of heating; hence the closed heater in the last stage must be so constructed that it can withstand a water pressure somewhat higher than the boiler pressure. An evaporator condenser is placed in the feedwater line; and various pumps and traps are installed to handle the condensate from the heaters.

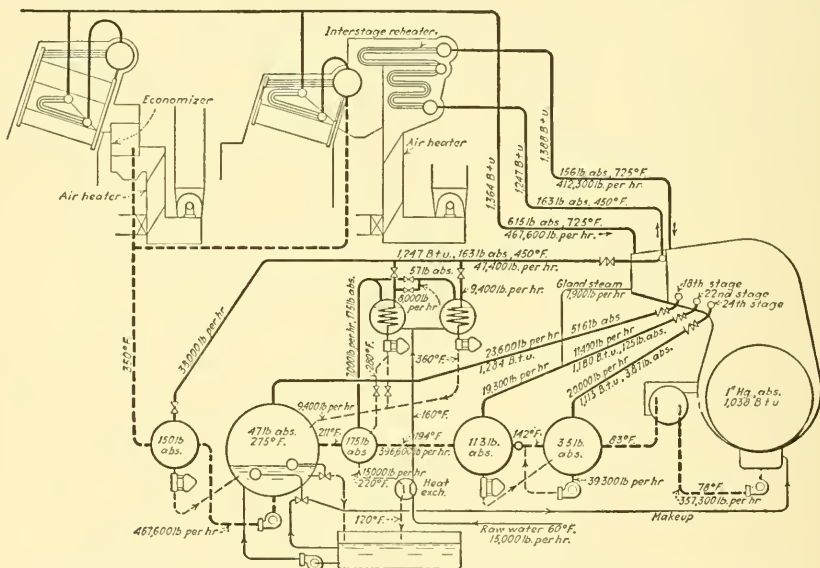


FIG. 988.—Flow Diagram of the Powerton Plant, Super Power Co. of Illinois.

(The net heat rate of this plant is 12,410 B.t.u. per kw-hr.)

(e) **Other notable installations** operating on the regenerative cycle include the State Line Station (see Fig. 351, p. 133) and the Lakeside Station (see Fig. 593, p. 509); recent additions, operating at a pressure of about 1300 lb. per sq. in., have been made to the latter plant.¹³ Other illustrative plant hook-ups, in most of which the regenerative cycle is employed, and in all of which the feedwater heating system

¹¹ For a complete description see *Power Plant Engineering*, July 15, 1930, p. 788.

¹² For a complete description see *Power*, June 10, 1930, p. 896.

¹³ See *Power Plant Engineering*, August 15, 1931, p. 830.

plays a vital part in the production of the proper energy balance, will be found in the technical press.¹⁴

596. Drainage of Regenerative Heaters.—The removal of condensate from the closed heaters of a regenerative system plays an important part in effecting the proper energy balance; and this may be accomplished in many different ways, of which four typical cases will be considered. In all except the third method it is assumed that the main condensate pump discharges through low-pressure and intermediate-pressure heaters to the boiler feed pump which, in turn, forces the water through a high-pressure heater to the boiler; in the third method the main condensate pump serves also as the boiler feed pump and forces the condensate through all of the heaters in series. (1) Each heater may have an independent drainage pump which delivers the condensate of that heater to the main boiler feed line on the boiler side of the heater. (2) The first arrangement may be modified by having each drainage pump deliver the heater condensate into the boiler feed line on the condenser side of the heater. As in the first arrangement, there are as many drainage pumps as there are heaters, which necessitates relatively high investment and maintenance costs; but in the second case the high-pressure drainage pump discharges to the *suction* side of the boiler feed pump and thus is not compelled to work against the full boiler pressure. Offsetting this advantage somewhat is the necessity of having a slightly larger boiler feed pump; however, this pump is more efficient than the small drainage pump. (3) Each of the higher pressure heaters may discharge its condensate through a trap into the next lower pressure heater wherein this water cascades and part “flashes” into steam. This is the simplest method of draining a heater; but it results in a small loss of available energy because of the irreversible expansion of the high pressure condensate, and it also prevents the use, in the lower pressure heater, of an equivalent amount of steam which has previously done useful work in the turbine; thus the efficiency of the cycle is somewhat reduced, as compared with cases (1) and (2). In this arrangement all heater condensate accumulates in the low-pressure heater from which it may be pumped or “flashed” through a trap, or its equivalent, into the main condensate line. (4) The condensate from the high-pressure heater may be cascaded into the

¹⁴ See the *Cochrane Handbook*, “Finding and Stopping Waste in Modern Boiler Rooms,” pp. 678 to 715; special summaries in *Power Plant Engineering*, in the Dec. 15th issues, 1925, 1928, 1930, and 1931; the issue of *Power* devoted to hook-ups, Nov. 25, 1930; “Extraction Heaters in the Power Plant,” by G. A. Gaffert, in *Power Plant Eng.*, April, 1933, p. 164 and August, 1933, p. 346.

intermediate-pressure heater, the combined condensate of the two being discharged from the latter into the main boiler feed line on the suction side of the feed pump. In this case the condensate from the low-pressure heater is allowed to flash into the condenser, or it is delivered to the suction side of the main condensate pump. The first two arrangements give the greatest degree of energy conservation, both being equal in this respect; the third system increases the energy consumption per kilowatt-hour by 40 to 50 B.t.u., as compared with (1) and (2); and the fourth method gives a result slightly better than the third.

597. Auxiliary Drives.—(a) **Steam turbines or engines** may be used alone, or in combination with electric motors, for driving auxiliaries, the proper arrangement being the one which gives the most economical and reliable results under the prevailing conditions. Such turbines or engines may receive their steam supply (1) direct from the boilers, (2) from a reduced-pressure auxiliary line, or (3) with steady-load operation, from one of the stages of the main turbines; and they may exhaust to a low-pressure stage of the main turbines, or to appropriate heaters.

(b) **Electric drives** for auxiliaries may obtain current from any of the following sources: (1) from the main-generator leads (through transformers, thus eliminating certain high-tension switching); (2) from the main bus, through transformers; (3) from an auxiliary generator mounted on the main-turbine shaft, thus taking advantage of the higher efficiency of the large main turbine as compared with that of a smaller one; (4) from a house turbine-generator; (5) from an outside source; or (6) from a combination of these sources, through such interconnections and with such transformations as are necessary. If direct current is used for the auxiliaries, it can be obtained from a direct-current house generator or from an alternating-current source through a motor-generator set.¹⁵

(c) The **total power required by electrically driven auxiliaries** depends to a large extent upon the type of fuel-burning equipment used, the station load factor, and the nature of the condensing water supply. According to L. W. Smith,¹⁶ the approximate percentage of the gross power produced which is required by all the electrically driven auxiliaries in a plant completely equipped with them, is 5.5 to 6.5 when pulverized coal is used, 4 to 5 with stoker firing, and 3.5 to 4

¹⁵ See the Symposium on Auxiliary Drive for Steam Power Stations presented to the A.I.E.E., Jan., 1932; Foreword, by F. H. Hollister; Steam Driven Auxiliaries, by W. P. Dryer; Electrically Driven Auxiliaries, by L. W. Smith (with a bibliography).

¹⁶ Ibid.

when the fuel is gas or oil. The subdivision of the auxiliary power in a modern 650-lb. steam power plant of 200,000-kw. capacity, burning pulverized coal and equipped entirely with electrically driven auxiliaries is, in per cent of the total auxiliary power, about as follows: Coal preparation, 22; boiler feed pumps, 22; turbine auxiliaries, 22; forced- and induced-draft fans, 20; other auxiliaries, 14; the total being 5.5 per cent of the gross power produced.

598. Conservation of Energy in Industrial Power Plants.—(a) Both process steam and electric energy are required in large amounts by many modern industrial plants, while steam for space heating is needed in most cases; and the proper method of obtaining these products with the greatest overall economy can only be decided after a careful analysis of the many factors involved. Usually, electric energy can be purchased from a public utility company, no matter what the geographical location of the plant may be; but the required supply of steam must, in almost all cases, be produced in the immediate vicinity of the plant by a steam-generating station designed especially to serve the needs of the given industry. If the proper relations exist between the demands for steam and electric energy, both of these demands may be met by a suitably designed steam power plant controlled by the industry itself; and various arrangements of this kind are employed in a wide variety of plants ranging from hotels, stores, office buildings, and city laundries, to those producing foods, beverages, paper, textiles, rubber products, and chemicals. Many of these plants are designed to generate steam at pressures comparable with those used in central stations.¹⁷

(b) The **energy-balance systems of industrial power plants** are frequently as complex as those used in central station practice, and are arranged with many variations to fit particular industrial needs. The prime movers in most of these systems are steam turbines, although piston engines are also used, especially with moderate steam pressures. However, in one recent plant, steam is generated at a pressure of 1800 lb. per sq. in. and is used directly in piston engines.¹⁸

The **proper source of steam** for process work or for space heating depends upon the variation in the demand and upon the pressure at which the supply is needed. Steam may be obtained from (1) the intermediate stages of an extraction turbine, (2) the exhaust of an engine or turbine, (3) evaporators, (4) accumulators, or (5) a combination of these. Since one of the major problems is the *effecting of a proper balance between the steam and electric demands under fluctuating*

¹⁷ See "High Pressure Industrial Plants in the United States and Canada," Power, May 28, 1929, p. 854.

¹⁸ See Power, March 18, 1930, p. 448; also May 19, 1931, p. 780.

loads, devices such as accumulators ¹⁹ may frequently be used to advantage; also if the process steam is needed at a moderate pressure, the exhaust from piston engines may be divided between the process line and a low-pressure or mixed-pressure turbine in proportion to the process demand,²⁰ the required turbine output being maintained by the admission of high-pressure steam to the first stage, when necessary. Evaporators may also be installed for the purpose of producing process steam at a desired moderate pressure, the heating coils being supplied with steam from the boilers, or from an extraction point or the exhaust of a high-pressure turbine (see Sect. 559(b)); and this arrangement has the advantage that the main power plant system forms an almost perfectly closed cycle, the necessarily large amount of make-up in the process system being handled independently of the working substance used in the boilers and turbine. Occasionally Diesel engines are included with steam-driven equipment in industrial power plants to provide for the generation of needed electric energy at times when the steam supply (which is governed by the process demand) is insufficient to allow the production of all this energy by the steam-driven prime movers.²¹

(c) **A determination of steam and power costs** must be made for each industrial power plant project before the necessity for the plant, or the kind and arrangement of equipment can be settled. This involves careful studies of the costs obtaining with the following combinations: (1) the purchase of both steam and electric energy; (2) the purchase of electric energy and the generation of steam in a plant owned by the industry; (3) the production of both steam and electric energy in a plant owned by the industry; (4) the production of both steam and electric energy in a plant operated by an independent organization for a specific industry; (5) the production of steam and electric energy in a plant having a large steam load and a small electric load, with the possibility of disposing of the surplus power advantageously to a public utility company; (6) the exchange of electric energy between industrial plants and public utility systems, process steam being produced in the former for private use. So many factors enter into the required studies that no attempt will be made to consider them further here; however, a large amount of literature on this general subject exists in the technical press, to which several references are given.²²

¹⁹ See Power, July 29, 1930, p. 168.

²⁰ See Power, May, 1933, p. 225.

²¹ See the yearly transactions of the Oil and Gas Power Division of the A.S.M.E.

²² See "Combined Heat and Power Supply in Industrial Plants," by W. F. Ryan,

599. Analysis of Energy-Balance Systems.—(a) The term **energy balancing** is used in power plant engineering in three ways: (1) it may be applied to the *energy balance sheet* which is drawn up to account in full for the utilization or rejection of the energy supplied to a machine, a group of machines, or a whole plant; (2) it may refer, in a steam-power plant, to the *act of maintaining a proper balance* between the energy made available for heating the feedwater (or supplying heat for buildings or for process work) and the amount that could be used with the greatest economy for such a purpose; and (3) it may be applied to the process of *making proper provision in the original design* of the plant for carrying out the balancing during subsequent operation. For each new plant, analyses should be made to determine which of the many possible *energy-balance systems* is the most feasible to use under the given conditions.

(b) An **energy-balance account** for a power plant is prepared upon the underlying principle that for each plant as a whole, and for each individual piece of equipment therein, the energy input must equal the energy output plus the losses. In the case of a steam power plant, interest may lie solely in the energy balance of the vapor cycle (or cycle of the working substance), that of the steam-generating system, that of an individual piece of apparatus, or that of the plant as a whole (i.e., the combination of the vapor cycle and the steam-generating system). For the *ideal vapor cycles*, the Rankine, regenerative, or reheating-regenerative, the methods of analyzing the energy balances are discussed in Part I, the first in Sect. 204, p. 321, and the other two in Chapter XVIII, p. 340.

(c) For an **existing plant** the energy-balance analysis can be readily

and "Engineering Aspects of Interchange of Power with Industrial Plants," by B. F. Wood, both presented at the annual meeting of the A.S.M.E., Dec., 1930; "Power and Heat in the Industrial Plant," by R. J. S. Pigott, Trans. A.S.M.E., FSP-52-38, 1930; special issue of Power, May 27, 1930; Power Plant Engineering, Jan. 1, 1931, p. 75; "Determining Steam Costs of Industrial Processes by Heat Consumption Method," by C. G. Klopp, Southern Power Journal, Jan., 1932, p. 17; "Calculating Steam and Power Costs in Industrial Plants," by H. L. Byrne, Power, Jan. 26, 1932, p. 140; "Cooperation between Private Plants and the Utilities," by O. H. Henschel, Power Plant Engineering, Feb. 15, March 15, April 15, and July 15, 1932, pp. 170, 255, 344, and 572; "An Analysis of Power and Heat Supply for Office Buildings," by L. E. Habben, Power, April 12, 1932, p. 550; "Design vs. Operation at the Hotel New Yorker," by W. D. Lewis, Power, May 3, 1932, p. 645; "Surplus Power from Industrial Plants," by A. G. Christie, Mechanical Engineering, Nov., 1932, p. 771; "Louisiana Steam Products Plant Serves Large Industry," in Power Plant Eng., Feb. 1, 1931, p. 182.

carried out if the necessary test data are available.²³ As a guide in the operation of a plant, a standard energy balance, or "bogey," may be prepared for the operating staff to follow.²⁴

(d) **For a proposed plant** the probable energy balance can be computed on the basis of assumed tabular data or curves for each piece of apparatus for the various loadings; such data should preferably be obtained from the guarantees of the manufacturers of the apparatus. The possible combinations of primary cycles, prime movers, steam generators, auxiliaries and their drives, heaters, evaporators, deaerators, economizers, air heaters, and other heat exchangers are so numerous and the conditions encountered are so variable that the subject can be covered herein only in a very brief and general way. The object of the energy-balance system is to so correlate the apparatus and its functioning as to best conserve the utilization of energy under all rates of steam generation, but particularly under the most representative rate; and for each set of conditions there is a definite feedwater temperature which will result in the best economy. Usually several analyses are required to determine the best combination of apparatus and the best feedwater temperature for use in each case.

(e) **One method of procedure** is as follows:²⁵

Assuming the auxiliaries to be steam-driven and knowing the fuel and air requirements of the steam-generating units, find the quantity of air handled by the induced- and forced-draft fans and the steam consumption of their engines or turbines; find the steam consumption of the various pumps (condenser and feedwater), and of any other auxiliaries; get the total amount of energy available from the exhaust steam for feedwater heating; then analyze the vapor cycle, and the effect of bleeding, by the following method.

The method of determining the effect of bleeding on the vapor cycle is one of

²³ The methods of determining and expressing the performance of steam power stations have been covered in Sect. 278, p. 120. The performance of steam-generating units is discussed in Sect. 396, p. 441. In carrying out the analysis it is necessary to know also the energy output of the main generators and the energy used by the auxiliaries; and for the prime movers, auxiliaries, and heaters, it is necessary to know the rates of flow of water and steam, the condition of the steam, and the entering and leaving temperatures and pressures.

²⁴ See report on "Power Station Betterment," Proc. N.E.L.A., 1928, p. 1508. Comparative energy-balance data for 20 plants are given in N.E.L.A. Proc., 1925, p. 1225, et seq. Also see "Heat Balance Study," by F. H. Rosencrants, Proc. N.E.L.A., 1924, p. 1194.

²⁵ References: "A Method for Calculating Central Station Heat Balance," by E. B. Hyde, Jr., and M. A. Guigou, in *Power*, July 3, 1928, p. 12; Serial Report on Station Betterment, N.E.L.A. Proc., 1928, p. 1508.

must be installed, and sufficient capacity must always be ready to meet the maximum demand although much of the equipment may be idle most of the time. Fixed charges are paid continuously on the equipment even if it be idle; only the apparatus which is fully loaded has the maximum earning capacity. Obviously, the character of the load demand, as shown by load curves, affects greatly the economic considerations entering into the selection and operation of plant equipment. After the equipment has once been installed, the proper division of the load between the various units to meet the different demands must be given careful consideration if the most economical operation is to be obtained; and this proper division is particularly essential if some of the units give better performance than others and if the units are of various sizes. The subject of plant economics is of very great importance, but it cannot be treated adequately except in considerable detail and at great length, space for which is not available in this text. In connection with such economic studies, however, and also in ordinary plant analyses, certain ratios and terms are employed, the meanings of which should be understood by all plant engineers. The more important of these will be given in the next section.

601. Load Curves; Definitions.—(a) The **types of load curves** used to show the variation in output of power plants during a given period of time include *daily load curves* (i.e., curves for each individual day), and *average 24-hour load curves* for the week, month, season, or year. Typical daily load curves for different kinds of loading are shown in Figs. 989 and 990, the former set being for various single loads and the latter for several arrangements of combined loads. These sets of curves illustrate the effect of different kinds of loads upon the variation in demand on the power plant; and they also show how the extreme variations in demand may be reduced, and the plant load factors increased, through the handling of combined loads rather than single ones. The daily load curve shown in Fig. 991 illustrates the effect (point X) of the sudden increase in lighting load caused by a heavy cloud obscuring a city area at mid-afternoon. Such an emergency may seriously tax the ability of the plant operators to put reserve equipment in operation fast enough to meet the sudden unexpected demand; but if a plant is to give “service,” it must always be ready to meet such conditions, even though a larger investment and an added operating expense may be involved.

(b) **Definitions** of terms and factors commonly used in connection with power generation are as follows:

(1) The **maximum demand** or **peak load** is the greatest demand appearing during the period under consideration.

(2) The **connected load** is the total rated capacity of all the energy-consuming apparatus connected to the system.

(3) The **demand factor** is the ratio of the maximum power demand to the total connected load.

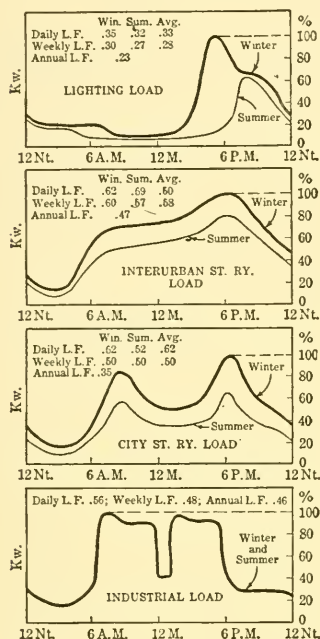


FIG. 989.—Typical Load Curves for Single Loads.

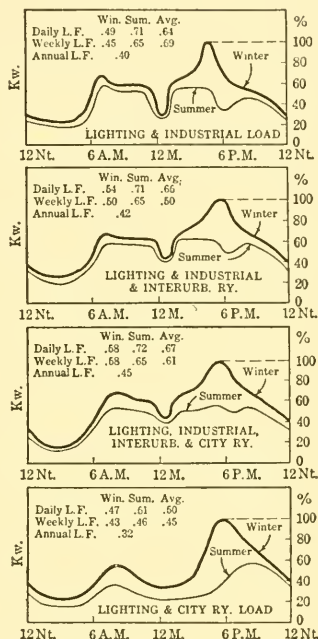


FIG. 990.—Typical Load Curves for Combined Loads.

(4) The **load factor** is the ratio of the average power produced to the maximum demand or peak load during the period under consideration.

(5) The **diversity factor** is the ratio of the sum of the maximum demands of various subdivisions of a system to the maximum demand of the combination of these subdivisions when functioning together.

(6) The **capacity factor** or **plant factor** is the ratio of the average load to the total rated capacity of the equipment installed to carry that load.

(7) The **installed capacity** is dictated by the maximum demand on the plant and includes also the capacity of the reserve units. By interconnection with other plants the capacity requirements may be reduced and the capacity factor for the combination increased, due to the better diversity factor obtained.

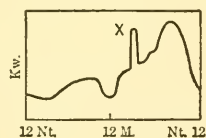


FIG. 991.

(8) The **total cost** of the power plant, often termed the "first cost," is made up of the initial expenditures for the land, building, equipment, installation, transportation, engineering fees, contractors' profits, and numerous other items connected with the construction of the plant.

(9) **Fixed charges** are the annual payments which must be met regardless of the operation or non-operation of the plant; they are *fixed* because normally they do not vary from year to year and are a definite percentage of the first cost of the plant. The principal items (with illustrative percentages) making up the fixed charges are: (a) *interest* on the capital invested (6 per cent); (b) allowance for amortization, obsolescence, or depreciation, this allowance usually being known simply as "*depreciation*" (5 per cent); (c) *taxes* and *insurance* (2 per cent up); (d) *maintenance* expense (2 per cent) if not handled as an operating charge. The total percentage is often taken as 15.

(10) **Justified additional investment** is that which involves lower fixed charges than the saving in annual operating expenses resulting from the extra investment.

602. Cost of Steam Power Plants.—(a) The **total cost of a power plant** is made up of items which vary widely in amount among plants; and it is influenced to a large extent by local conditions, the character of the plant loading, and the ideas back of the design. In general, the cost of a completed plant per kw. of installed capacity is higher for small plants than for large ones; it may be a little greater for high-pressure and high-temperature plants than for ones using more moderate temperatures and pressures, and is justly more for heavily loaded base-load plants than for plants having light variable loads on the average. Often a higher first cost may be justified by the better performance resulting therefrom. In view of all of the variables involved, it is not possible to present briefly cost data which can be used in any way except as being merely approximate or illustrative; and the values which follow should be so used.

(b) The **cost of a completed steam plant per kw. of installed capacity** ranges from \$60 to \$250 or more; some more specific values will appear later. In comparison, corresponding unit costs for Diesel plants range from \$125 to \$350; and for hydraulic plants the range is from \$100 to \$500 or more, depending largely on the cost of the developments necessary to make the water power available.

Based on data collected from a group of representative modern central stations (1928), the unit investment costs in dollars *per kva.* for the plants and their component parts were found by L. W. W. Morrow²⁶ to have the wide range of values given in Table LXXXIV.

²⁶ "Power Costs in Large Plants," *Electrical World*, Oct. 27, 1928, p. 827.

TABLE LXXXIV
 RANGE OF UNIT INVESTMENT COSTS OF CENTRAL STATIONS
 (In Dollars per Kilovolt-Ampere)

	Maximum	Minimum	Average	Range
Total station cost, including land.....	144	81	114.0	1.7 : 1
Total cost of building and equipment....	140	65	102.0	2.3 : 1
Total building cost.....	52	19	32.0	2.7 : 1
Foundations and substructure cost....	21	4	10.6	5.2 : 1
Superstructure cost.....	40	10	22.0	4 : 1
Total equipment cost.....	101	46	70.0	2.2 : 1
Boilers and accessories.....	30	11	21.0	2.7 : 1
Station piping.....	10	3	7.0	3.3 : 1
Turbo-generators and condensers.....	28	13	22.0	2.1 : 1
Electrical equipment in switch house..	13	6	8.5	2.1 : 1

Morrow suggests \$77 *per kva.* as the goal to be sought for the unit cost of a complete central station. The figures given can be reduced to the basis of unit costs *per kw.* of installed capacity by dividing them by the plant power factor, which commonly has a value of about 80 per cent.

As the **unit volumes of plant buildings** have an important influence on the cost of the plants, Table LXXXV, giving the range of unit volumes of these same stations is of interest:

TABLE LXXXV
 UNIT VOLUMES OF CENTRAL STATIONS
 (Cubic Feet per Kilovolt-Ampere)

	Maximum	Minimum	Average	Range
Coal preparation and handling.....	10	2.2	6	4.5 : 1
Boiler room.....	50	14.0	21	3.5 : 1
Turbine room.....	35	4.5	15	7.8 : 1
Switch house.....	27	0.6	6	45 : 1

(c) The **distribution of costs** among the main subdivisions of stations is illustrated on the unit cost basis by the data given in Table LXXXIV, and on the percentage basis by Figs. 992 and 993.

²⁷ See Power, April 24, 1928, p. 714.

(d) **High-pressure plants** do not necessarily cost more than low-pressure stations of the same capacity, as the decreased size of the equipment largely offsets the increased costs that would otherwise occur through the use of high pressures.

The relative costs of a 400-lb. and a 1400-lb. installation, and the relative amounts of space occupied per kw., are given ²⁸ in Tables LXXXVI and LXXXVII respectively. In general it may be stated that a 1400-lb. plant can be built for approximately the same total unit cost as a 400-lb. plant; for, although the steam generators and station piping will have higher unit costs in the former plant, many of the boiler-room auxiliaries will show a considerably lower value, and while the unit cost of the turbines may be slightly greater, that of the building and the condensing equipment may be materially less, in the

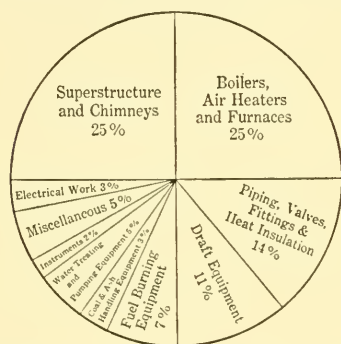


FIG. 992.—Cost Distribution in a Heating Plant.

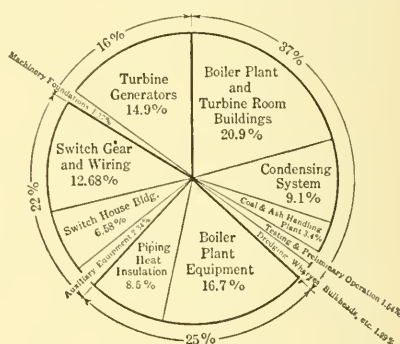


FIG. 993.—Cost Distribution in a 350-Lb. Plant.²⁷

1400-lb. installation. The unit cost of the electrical equipment is not affected appreciably by the magnitude of the pressure.

(e) **Detailed costs** of two small plants ²⁹ are given in Table LXXXVIII. They are based on sample designs prepared by a committee of the N.E.L.A. and on Philadelphia prices as of 1924. The 1500-kw. plant (Fig. 981) has four 500-kw. generating units, three for normal operation and one for reserve; the 7500-kw. plant (Fig. 982) has four 2500-kw. units, including one spare. The values tabulated are based upon the total installation costs, including engineering fees, contractors' profits, and contingencies. The operating conditions and characteristics are appended at the end of the table.

The costs of industrial steam-generating equipment ³⁰ are given in

²⁸ Proceedings N.E.L.A., 1929, p. 1210.

²⁹ Proc. N.E.L.A., 1924: "The Design and Maintenance of a 1500-Kw. Central Station Plant," p. 1000; similar but for a 7500-kw. plant, p. 1413.

³⁰ From "Industrial Boiler Plant Design," by H. Blietfret, Power, Oct. 6, 1931, p. 493.

TABLE LXXXVI

RELATIVE COSTS OF 400-LB. AND 1400-LB. INSTALLATIONS—STEVENS & WOOD

	Cost—Dollars per Kw.	
	Toronto, 140,000 Kw. 400 Lb. Pressure	Deepwater, 159,000 Kw. 1400 Lb. Pressure
Land.....	\$0.82	\$1.57
Preparation of site.....	1.64	0.21
Yard work.....	4.42	4.48
River front work.....		2.68
Foundations.....	15.93	4.16
Building.....	10.78	10.11
Piping (not including circulating water).....	4.75	5.09
Boiler-room equipment.....	23.41	31.01
Turbine-room equipment.....	26.47	21.34
Electrical switching equipment (in station).....	12.11	8.98
Service equipment.....	2.43	5.37
Coal-handling.....	4.67	3.19
Step-up substation.....	6.68	6.62
Preliminary operation.....	0.74	0.50
Total.....	\$113.85	\$105.31

TABLE LXXXVII

	Toronto	Deepwater
Plan area, sq. ft. per kw.....	0.49	0.40
Volume above basement floor, cu. ft. per kw.....	63.0	40.00

Table LXXXIX; and the volume, structural steel weights, and cost of the buildings with coal bunkers for these plants are given in Table XC.

(f) The **cost of equipment** generally varies approximately according to the equation $y = b + mx$, in which b represents a fixed amount to cover such manufacturing cost items as overhead expense and allowances for an increasing ratio of weight to capacity with a decrease in the size of the apparatus, m is an additional cost per unit of size, and x is the size expressed in customary units, such as kilowatts or square feet of surface. *Selling prices* depend on the cost of labor, materials, and

TABLE LXXXVIII

TYPICAL COSTS OF 1500-Kw. AND 7500-Kw. CENTRAL STATIONS
(Installed capacity, 2000 kw. and 10,000 kw., respectively)

(Cost per kw., installed, including spare units)

Total kw. installed, including one spare unit.....	2000	10,000
Real estate.....	\$4.00	\$1.74
Building.....	68.90	42.70
Railroad siding.....	1.27	0.33
Boilers, settings, and superheaters.....	19.00	10.20
Stokers.....	5.43	4.41
Condensers.....	11.45	7.77
Piping.....	29.00	11.59
Turbine-generators (4 units).....	41.25	24.50
Switchboard.....	8.80	3.37
Wiring (motor and light).....	6.60	2.61
Outside distributing structure.....		1.14
Chimney and flue.....	9.00	2.62
Forced-draft fans and ducts.....	3.00	1.35
Ash and coal-handling equipment.....	17.13	5.80
Feedwater heaters.....	0.99	0.97
Feed pumps.....	2.00	0.60
House service pump and surge tank.....		0.31
Air compressor.....		0.18
Equipment foundations.....	4.11	2.66
Cooling water supply.....	5.53	5.24
Water softener.....		0.30
Generator ventilating ducts.....		1.31
Engine-room crane.....	1.05	0.25
Machine-shop equipment.....		0.93
Oil filters and tanks.....	1.25	0.35
Total cost per kw. installed.....	\$239.77	\$133.25

Plant	Units	Steam Characteristics
1500 kw. (Peak)	4-500 kw. (Total = 2000 kw.)	Pres., 200 lb./sq. in. gauge; superheat, 150 deg. fahr.; exhaust pressure, 1 lb./sq. in. abs.
7500 kw. (Peak)	4-2500 kw. (Total = 10,000 kw.)	Pres., 200 lb./sq. in. gauge; superheat, 162 deg. fahr.; exhaust pressure, $\frac{3}{4}$ lb./sq. in. abs. No extraction feedwater heating.

Coal—\$7.00 per 2000 lb. delivered; 13,500 B.t.u. per lb.

Efficiency of boiler and furnace: 65 per cent at 25 per cent yearly load factor
70 per cent at 50 per cent yearly load factor
73 per cent at 75 per cent yearly load factor

TABLE LXXXIX

ANALYSIS OF COSTS FOR INDUSTRIAL BOILER PLANTS

(In Dollars per 10 Sq. Ft. of Boiler Heating Surface)

Foundations for building equipment.....	8 to 12
Building and fuel bunkers.....	20 to 40
Boilers, superheaters, soot blowers (erected) *.....	20 to 35
Economizer (erected) *.....	10 to 20
Air heaters (erected).....	10 to 15
Settings.....	6 to 10
Coal firing and forced-draft equipment †.....	12 to 18
Coal-handling equipment.....	5 to 7
Ash-handling equipment.....	4 to 5
Breeching and chimney.....	7 to 10
Feed pumps.....	2 to 3
Piping.....	10 to 15
Water-treating plant.....	6 to 9
Motors, wiring and lighting.....	6 to 8
Combustion control.....	1 to 3

* Boiler pressures of 250 lb. per sq. in. and less.

† Including unit mills in case of powdered coal.

TABLE XC

VOLUME, STRUCTURAL STEEL WEIGHTS, AND COST OF BOILER HOUSES WITH COAL BUNKERS

	Building Space, Cu. Ft. per 10 Sq. Ft. of Boiler Surface	Weight of Structural Steel, Lb. per 10 Sq. Ft. of Boiler Surface	Cost of Building and Coal Bunkers above Foundation	
			Per Cu. Ft. of Building Space	Per 10 Sq. Ft. of Boiler Surface
Two 5120-sq. ft. straight-tube boilers, 3 unit mills, overhead bunkers.....	118	197	\$0.22	\$26.50
Four 6000-sq. ft. 4-drum Stir- ling-type boilers, traveling grate stokers, overhead bunkers.....	158	280	0.25	39.30
Six 8140-sq. ft. 3-drum Stirling- type boilers, traveling grate stokers and blast furnace gas, overhead bunkers.....	150	270	0.16	23.55
Six 9120-sq. ft. boilers, econ- omizers, 8 unit mills, 1 travel- ing grate stoker, overhead bunkers.....	193	269	0.17	31.95

* Based on Fig. 327, p. 96. Delivered and erected without auxiliaries; steam pressures less than 400 lb./sq. in.; steam temp. < 725 F; vacuum $\bar{\geq}$ 29 in. Hg, if condensing.
† Solid-forged drums.

transportation, on competition between sellers and between buyers, on the condition of the money market, and on many other items. Illustrative costs of equipment are given in Table XCI, but these values are merely approximate for the reasons already given. Other examples of the cost of equipment and of its installation in power plants will be found in the tables and figures given in the preceding parts of this section. The final cost analysis for an actual plant should be based on manufacturers' quotations for the particular apparatus under consideration.

603. Cost of Energy.—(a) The total cost of energy per kw-hr. at the station *bus bars* is the sum of (1) the fixed charges, (2) the operating expenses, and (3) the maintenance expense. The **fixed charges** per kw-hr. are found by dividing the annual fixed charge by the total kw-hr. output per year. The **operating expense** is made up of the payments for fuel, labor, and supplies. The **maintenance expense** includes all expenditures for labor, new parts, and material required for keeping the equipment in good running order. The sum of the operating and maintenance expenses is termed the **production cost**. The three numbered items, the production cost, and the total cost, all per kw-hr., for the typical central stations already considered in Table LXXXIV, are given in Table XCII, to which has been added the number of men required per day for operating 1000 kva. of station capacity.

TABLE XCII

UNIT PRODUCTION COSTS IN CENTS PER KILOWATT-HOUR

	Maximum	Minimum	Average	Range
Operating cost, total.....	0.489	0.189	0.322	2.6 : 1
Fuel.....	0.418	0.142	0.240	2.9 : 1
Wages and superintendence.....	0.115	0.031	0.067	3.7 : 1
Water, lubricants, and supplies.....	0.016	0.003	0.009	5.3 : 1
Miscellaneous.....	0.041	0.001	0.012	4.1 : 1
Maintenance cost, total.....	0.056	0.018	0.038	3.1 : 1
Buildings.....	0.009	0.0007	0.0052	1.3 : 1
Boilers and accessories.....	0.041	0.006	0.022	6.8 : 1
Prime movers and accessories.....	0.022	0.001	0.0062	2.2 : 1
Generators and electrical equipment..	0.01	0.001	0.0034	10 : 1
Production cost, total.....	0.545	0.207	0.36	2.7 : 1
Fixed charge, total.....	0.895	0.146	0.50	6 : 1
Total cost of energy.....	1.44	0.353	0.86	4 : 1
Number of men per 1000 kva.....	1.8	0.5	1.2	3.6 : 1

TABLE XCIII
INVESTMENT AND OPERATING COSTS OF 1500-KW. CENTRAL STATIONS.
INVESTMENT COSTS

Kind of Plant	Turbine	Uniflow Engine	Simple Four-Valve Engine	Compound Four-Valve Engine	Simple Corliss Engine	Compound Corliss Engine	Diesel Engine
Real estate	\$8,000	\$8,000	\$8,000	\$8,000	\$8,000	\$8,000	\$8,000
Building	137,800	187,200	158,400	161,600	164,000	173,200	107,200
Boilers and settings	38,000	37,100	52,300	37,100	52,300	37,100	
Stokers	10,850	10,850	15,920	10,840	15,920	10,850	
Condensers	22,900	11,880	14,550	11,470	15,050	11,880	
Piping	58,000	58,000	58,000	58,000	58,000	58,000	19,000
Generating units	82,500	135,700	131,500	142,500	131,500	158,500	321,000
Switchboard	17,600	17,600	17,600	17,600	17,600	17,600	17,600
Wiring	13,200	13,200	13,200	13,200	13,200	13,200	13,200
Chimney and flue	18,000	18,000	18,000	18,000	18,000	18,000	
Forced-draft fans and ducts	6,000	6,000	6,500	6,000	6,500	6,000	
Coal- and ash-handling equipment	34,260	34,260	35,250	34,260	35,250	34,260	
Engine-room crane	2,100	2,550	2,400	2,400	2,550	2,550	2,550
Feedwater heater	1,980	1,980	2,500	1,980	2,500	1,980	
Feed pumps	4,010	4,010	4,010	4,010	4,300	4,010	
Equipment foundations	8,230	22,000	15,800	22,375	16,860	19,950	13,700
Oil filters and tanks	2,500	2,500	2,500	2,500	2,500	2,500	
Fuel oil storage	11,060	6,470	6,800	6,470	6,800	6,470	16,100
Railroad siding	2,550	2,550	2,550	2,550	2,550	2,550	9,680
Total	\$479,540	\$579,850	\$565,780	\$560,855	\$573,380	\$586,600	\$530,580
Cost per kw. (1500 kw. Peak)	\$319.69	\$386.56	\$377.19	\$373.90	\$382.25	\$391.07	\$353.72
(2000 kw. installed capacity)							

OPERATING COSTS

	Load Factor 25 Per Cent						
Fuel	\$53,655	\$51,100	\$59,850	\$50,610	\$63,700	\$51,450	\$31,200
Labor	11,580	11,580	11,580	11,580	11,580	11,580	12,600
Lubricants	500	500	500	500	500	500	1,000
Supplies	2,800	3,300	3,400	3,300	3,500	3,300	6,800
Cooling water							800
Operation (total)	\$68,285	\$66,480	\$73,330	\$65,990	\$79,280	\$66,830	\$52,400
Maintenance	7,000	8,000	7,500	7,500	7,500	7,500	8,000
Production (total)	\$75,285	\$74,480	\$82,830	\$73,490	\$86,780	\$74,330	\$60,400
Fixed charges at 15 per cent	71,931	86,978	84,867	84,128	86,007	87,990	79,587
Total cost at bus.	\$147,216	\$161,458	\$167,697	\$157,618	\$172,787	\$162,320	\$139,987
Cost per kw-hr.							
Production	\$0.0229	\$0.0227	\$0.0252	\$0.0224	\$0.0265	\$0.0226	\$0.0184
Fixed charges	0.0219	0.0265	0.0258	0.0256	0.0262	0.0268	0.0242
Total	\$0.0448	\$0.0492	\$0.0510	\$0.0480	\$0.0527	\$0.0494	\$0.0426

Load Factor 50 Per Cent

Fuel	\$78,400	\$78,750	\$94,500	\$78,225	\$102,900	\$83,300	\$54,750
Labor	12,660	12,660	12,660	12,660	12,660	12,660	12,600
Lubricants	300	550	550	550	550	550	1,200
Supplies	3,000	3,500	3,600	3,500	3,700	3,500	7,200
Cooling water							1,000
Operation (total)	\$94,360	\$95,460	\$111,310	\$94,935	\$119,810	\$100,010	\$76,750
Maintenance	9,850	10,800	10,000	10,000	10,000	10,000	10,500
Production (total)	\$104,210	\$106,260	\$121,310	\$104,935	\$129,810	\$110,010	\$87,250
Fixed charges at 15 per cent	71,931	86,978	84,867	84,128	86,007	87,990	79,587
Total cost at bus.	\$176,141	\$193,238	\$206,177	\$189,063	\$215,817	\$198,000	\$166,837
Cost per kw-hr.							
Production	\$0.0158	\$0.0161	\$0.0185	\$0.0160	\$0.0197	\$0.0168	\$0.0133
Fixed charges	0.0110	0.0133	0.0129	0.0128	0.0131	0.0134	0.0121
Total	\$0.0268	\$0.0294	\$0.0314	\$0.0288	\$0.0328	\$0.0302	\$0.0254

Load Factor 75 Per cent

Fuel	\$99,750	\$100,275	\$123,200	\$99,750	\$135,975	\$107,800	\$78,450
Labor	12,660	12,660	12,660	12,660	12,660	12,660	13,680
Lubricants	350	600	600	600	600	600	1,400
Supplies	3,200	3,700	3,800	3,700	3,900	3,700	7,600
Cooling water							1,200
Operation (total)	\$115,960	\$117,235	\$140,260	\$116,710	\$153,135	\$124,760	\$102,330
Maintenance	11,000	12,000	11,500	11,500	11,500	11,500	12,000
Production (total)	\$126,960	\$129,235	\$151,760	\$128,210	\$164,635	\$136,260	\$114,330
Fixed charges at 15 per cent	71,931	86,978	84,867	84,128	86,007	87,990	79,587
Total cost at bus.	\$198,891	\$216,213	\$236,627	\$212,338	\$250,642	\$224,250	\$193,917
Cost per kw-hr.							
Production	\$0.01290	\$0.01317	\$0.01538	\$0.01296	\$0.01667	\$0.01387	\$0.01160
Fixed charges	0.00730	0.00883	0.00862	0.00854	0.00873	0.00893	0.00808
Total	\$0.0202	\$0.0220	\$0.0240	\$0.0215	\$0.0254	\$0.0228	\$0.01968

Steam Pressures:

Turbine, 200 lb. per sq. in. gauge.

Uniflow and four-valve, 175 lb. per sq. in. gauge

Corliss, 150 lb. per sq. in. gauge.

Coal: 13,500 B.t.u. per lb.; \$7.00 per net ton delivered.

Oil for Diesel engine: 18,500 B.t.u. per lb.; \$3.00 per bbl.

Vacuum referred to a

30-in. barom.

Turbine, 28 in. Hg.

Engine, 26 in. Hg.

All steam units are

operated condensing.

Superheat:

Turbine, 150° F.

Uniflow, 150° F.

Four-valve and Cor-

liss, 100° F.

(b) **Comparative costs for a 1500-kw. power plant** (Figs. 980 and 981), as designed with seven different types of prime movers, are given in Table XCIII;³¹ in addition to investment costs, the table also gives both operating and total unit costs for each of three yearly load factors; thus a comprehensive comparative study may be made. A résumé of the costs for the *turbine plant* given in this table is arranged in Table XCIV,

TABLE XCIV
INVESTMENT AND OPERATING COSTS PER KW-HR. GENERATED
(1500- and 7500-kw. turbine plants)

Load Factor	25 Per Cent		50 Per Cent		75 Per Cent	
Plant size, kw.	1500	7500	1500	7500	1500	7500
Fuel.....	\$0.0164	\$0.0124	\$0.0120	\$0.00877	\$0.01015	\$0.00697
Labor.....	0.0035	0.0011	0.00193	0.00071	0.00129	0.00055
Supplies.....	0.00086	0.0003	0.00045	0.00019	0.00033	0.00016
Maintenance.....	0.00214	0.00122	0.00150	0.00107	0.00112	0.00091
Fixed charges @ 15 per cent.	0.0219	0.01217	0.0110	0.00608	0.00732	0.00406
Total.....	\$0.0448	\$0.02727	\$0.0268	\$0.01685	\$0.0202	\$0.01267

and these costs may be compared directly, in the same table, with the corresponding ones for a 7500-kw. turbine plant (Fig. 982) of similar design (also see Table LXXXVIII). The labor costs for these two turbine plants were assumed to be made up according to Table XCV.

(c) The **cost of energy to the consumer** per kw-hr., not only includes the cost at the plant, but also the fixed charges and maintenance expense for the distributing system, the cost of meter reading, the office and administration charges, the reserve, and the profits. A typical distribution of the expense items making up the total cost to the consumer, is given in Table XCVI as an average for companies serving urban areas where a density of 2000 customers per square mile is found.³² It should be noted that, in this table, the sum of the fixed charges and item 7 makes up eight-tenths of the total cost at the customer's meter.

When a plant serves but one nearby customer, the energy cost at the customer's meter is but little more than the cost at the station,

³¹ From the Proceedings of the N.E.L.A., 1924, p. 1001.

³² Power Plant Engineering, Aug. 1, 1932, p. 580.

TABLE XCV
OPERATING LABOR IN 1500- AND 7500-Kw. PLANTS

Load Factor	25 Per Cent		50 Per Cent		75 Per Cent		Salaries per month	
	1500	7500	1500	7500	1500	7500	1500	7500
Plant size, kw.....								
Chief engineer.....	1	1	1	1	1	1	\$225	\$250
Watch engineers.....	2	3	2	3	2	3	150	175
Assistant engineers.....	0	1	0	2	0	3	0	125
Electricians.....	0	1	0	2	0	3	0	150
Firemen.....	3	3	3	3	3	4	125	125
Laborers.....	1	1	2	2	2	3	90	84
Total.....	7	10	8	13	8	17	\$590	\$909

TABLE XCVI
ENERGY COST PER Kw-Hr. AT CUSTOMER'S METER

	Cents
1. Generating electricity.....	0.47
2. Fixed charges on the generating station system that produces electricity..	0.78
3. Transmission and substation operation.....	0.16
4. Fixed charges on transmission and substation equipment.....	0.33
5. Cost of operating distribution lines from the substation to the consumer's meter.....	0.29
6. Fixed charges on the distribution equipment.....	1.45
7. Reading meters, preparing bills, collections, service calls and business administration.....	1.80
8. Total cost of kilowatt hour, per unit of electric service, at consumer's meter..	5.28

since item 7 and the fixed charges on transmission, substation, and distribution equipment are largely eliminated; however, it should be noted that the *magnitude* of the cost per unit of energy delivered depends directly upon the yearly load factor, a high one making possible a lower charge for energy than is permissible with a low one. When the system serves many small customers through an expensive distributing network, there is a large added cost for additional fixed charges and for service, accounting, and administration.

CHAPTER L

VENTILATION, HEATING, COOLING, AND AIR CONDITIONING

604. General.—The properties of the air in an enclosure may be controlled to meet the physiological requirements of human comfort or to assist the carrying out of certain industrial processes. Ventilation, heating, cooling, and air conditioning are processes that control some property or properties of the air. Some of the principal factors that affect human comfort as influenced by air environment are the (1) dry-bulb temperature, (2) humidity, (3) motion, (4) distribution, (5) dust content, (6) bacteria content, and (7) odors of the air. Other factors that may influence comfort, but the effects of which are not so well established at the present time, are (8) light, (9) ozone content, (10) ionic content, and (11) pressure. While the processes of *heating* and *cooling* fundamentally control only temperature, and while *ventilation* means the process of changing the air within an enclosure by supplying fresh air and exhausting used air, the process of *air conditioning* is now understood to mean the simultaneous control of at least (1), (2), and (3) of the factors listed above.

605. Human Comfort and Air Environment.—(a) Human occupancy of a confined space produces at least six important alterations in the properties of the air: (1) oxygen is used in metabolism (the combustion of food) and the oxygen content of the confined air is thereby decreased slightly; (2) carbon dioxide, a waste product of metabolism, is exhaled and the carbon dioxide content of the air is thereby increased; (3) products of organic decomposition are given off; (4) the temperature of the air is raised, because the energy liberated in metabolism is largely lost as heat from the body surface; (5) the humidity of the air is increased by the evaporation of moisture from the respiratory tract and the body surface; and (6) the number of positive and negative ions in a unit volume of the air is decreased.

The old and fallacious beliefs were that human discomfort was due either to the increase in the carbon dioxide content of the air in the enclosure, or to a poisonous effect of the products of organic decomposition in respired air. Many experiments have shown the fallacy of these beliefs. Subjects have been placed in air-tight chambers until the

temperature, humidity, and carbon dioxide content of the enclosed air increased appreciably and discomfort resulted, while other subjects outside of the chamber breathing the air of the chamber through tubes, but with their bodies exposed to normal air conditions, suffered no discomfort. Experiments¹ conducted in rooms ventilated by both natural and mechanical means indicate that the ionic content of the air decreases due to human occupancy. Prohibitive rates of outdoor air supply are necessary to maintain the normal ionic content, but artificial ionization is possible. Whether this decrease in ionic content influences human comfort or health has not yet been conclusively demonstrated.

It is now believed that human discomfort results when the tempera-

ture-regulating system of the body is unable to compensate comfortably for differences between the rate of energy release by metabolism and the rate of energy dissipation from the body by evaporation and heat transfer.

The rate of heat loss from the body by evaporation and heat transfer depends upon the body temperature and upon the dry-bulb temperature, the partial pressure of the contained water vapor, and the velocity of the surrounding air. Obviously, neither dry-bulb temperature nor wet-bulb temperature, alone, is a complete index of human comfort.

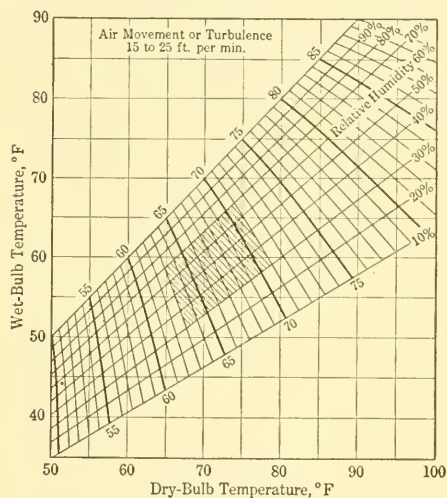


FIG. 994.—Effective Temperatures and Winter Comfort Zone. (Subjects at Rest in Still Air.)

(b) A consistent relationship exists between various combinations of temperature, humidity, and air motion and their effect on a person's feeling of warmth and the physiological effects on the body, according to experiments conducted at the Research Laboratory of the A.S.H. and V.E. and at the Harvard School of Public Health.² Points representing conditions of equal warmth, when spotted on a coordinate system of wet-bulb temperature vs. dry-bulb temperature (for any constant air velocity), fall on smooth curves as shown in Fig. 994.

¹ "Changes in Ionic Content of Air in Occupied Rooms," by Yaglou, Benjamin, and Choate, in *Heating, Piping, and Air Conditioning*, 1931, p. 865.

² Reported in *Trans. A.S.H. and V.E.*, Vols. 27 to 36 inc.

These lines of constant feeling of warmth are called lines of constant **effective temperature**, and the numerical values of the effective temperatures are those of the dry-bulb (also wet-bulb) temperature at which the lines of equal warmth cross the line of 100 per cent relative humidity for "still air conditions" (air velocities from 15 to 25 ft. per min.). Strictly, effective temperature is not a temperature at all, but an arbitrary index of the degree of warmth or cold felt by the normal human body in response to the composite effects of dry-bulb temperature, humidity, and air movement.

(c) Also in Fig. 994 is shown the **average winter comfort zone** for

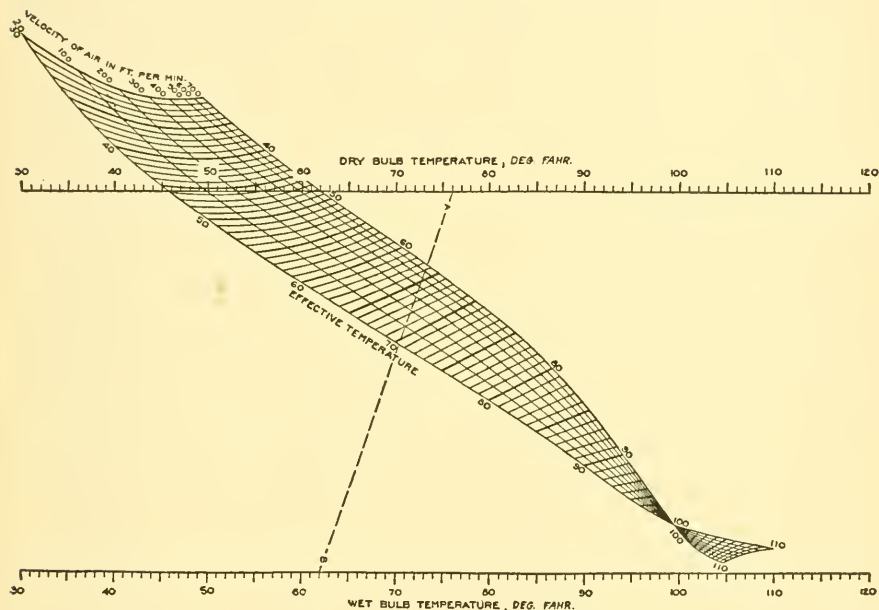


FIG. 995.—Effective Temperatures for Persons Normally Clothed.

persons normally clothed and at rest in still air indoors. This zone includes those air conditions in which 50 per cent or more of the experimental subjects were comfortable, and the effective temperature limits of the zone are 63 and 71, while the arbitrary limits placed upon relative humidity are 30 per cent and 70 per cent. Thus, for an indoor dry-bulb temperature of 70 deg. fahr., and an indoor wet-bulb temperature of 55 deg. fahr., the effective temperature is about 65 deg., the relative humidity is about 37 per cent, and the point falls in the average winter comfort zone.

(d) The **average summer comfort zone**, for exposures to the artificially conditioned air of 3 hours or more, appears to range from 67 to

75 deg. effective temperature. This increase over the winter comfort zone is probably due partly to the difference in clothing and partly to the adaptation to seasonal weather. Experience has shown that it may be undesirable to maintain too great a difference between the outside and inside dry-bulb temperatures, or too low an effective temperature, particularly for short exposures to an artificially cooled enclosure; e.g., for an outside dry-bulb temperature of 100 deg. fahr., an indoor dry-bulb temperature of, say, 82 deg. fahr. and an indoor wet-bulb temperature of, say, 67 deg. fahr. (to give an effective temperature of 75) would prove much more comfortable than an indoor dry-bulb of 75 deg. fahr. and a wet-bulb of 60 deg. fahr. (to give an effective temperature of 69), even though the latter is within the average summer comfort zone.

The combined effect of air velocity, dry-bulb temperature, and wet-bulb temperature upon effective temperature is shown in the alignment

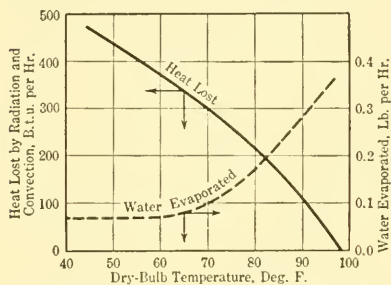


FIG. 996.—Heat and Moisture Loss from Average Man at Rest in Still Air.

chart of Fig. 995, originally prepared by Yaglou and Miller.³ The example solved on the chart is for a dry-bulb temperature of 76 deg. fahr. and a wet-bulb temperature of 62 deg. fahr.; if the air velocity is 100 ft. per min., the effective temperature is 69, while if the air velocity is 0, the effective temperature is 70.4, and the reduction in effective temperature of 1.4 deg. is caused by the increase in air velocity.

(e) The amount of heat lost by radiation and convection from the surface of the body of an average man (19.5 sq. ft. of body surface) at rest in still air⁴ is shown in Fig. 996, as a function of the dry-bulb temperature of the air. Although the relative humidity of the air affects this relationship, this effect may be neglected in the usual applications of this information. In the same figure, the weight of water evaporated from the body surface and the respiratory tract is also shown as a function of the dry-bulb temperature; again, the effect of relative humidity upon this loss of weight is neglected.

As an example of the use of Fig. 996, find the amount of heat and water vapor added to the atmosphere of an auditorium by 1000 adults, seated at rest, if the dry-bulb temperature is 75 deg. fahr. From Fig. 996, the amount of heat lost by radiation and convection per hour per adult is 260 B.t.u.; the amount of heat added

³ Trans. A.S.H. and V.E., Vol. 31, 1925, p. 89.

⁴ For complete data, see Trans. A.S.H. and V.E., Vol. 35, 1929, p. 245.

per hour is 260,000 B.t.u. From the same figure, the weight of water vapor added is 0.13 lb. per hour per adult; the total weight of water vapor added per hour to the atmosphere is 130 lb.

606. Ventilation.— (a) The purpose of ventilation, in a restricted sense, is to produce a change of air. However, the quality or state of the air supplied is as important in good ventilation as the quantity. Sufficient air must be supplied and exhausted to remove (1) the heat and water vapor given off by occupants and the heat given off by mechanical equipment, (2) odors, and (3) injurious substances carried by the air, such as dust, fumes, smoke, and bacteria. At the same time, the rate of air supply and removal should not be so great as to produce uncomfortable air velocities or drafts within the enclosure.

(b) **Ventilation standards** fixing the **quantity** of air to be supplied are highly arbitrary. In fact, one simple, arbitrary, and sweeping standard, that has been used for many years and is specified in many state and federal codes, requires the supply of at least 30 cu. ft. of fresh air per minute per person. The volume of air supplied to a room in unit time may be determined by measuring the amount of carbon dioxide in the room. An absorption burette filled with a saturated solution of caustic potash is then used to find the CO₂ content at several points in the room. If each person be assumed to produce 0.01 cu. ft. of CO₂ per min., and if the outside air consists of 4 volumes of CO₂ in 10,000, then the volume of air being supplied to and exhausted from the room per occupant per minute, neglecting the dilution effect of the air initially filling the volume of the enclosure and the absorption of CO₂ by walls or other surfaces, is

$$V = \frac{100}{\text{CO}_2 - 4} \quad . \quad . \quad . \quad . \quad . \quad . \quad (683)$$

where V = the volume of fresh air supplied and exhausted per occupant, in cu. ft. per min.

CO₂ = carbon dioxide content of room air, volumes in 10,000.

(c) One arbitrary **standard of quality** of ventilation is the Synthetic Air Test and Chart devised by Dr. E. V. Hill. The factors affecting the quality of the ventilation that are measured and recorded in this test are the effective temperature, dust particles, bacteria colonies, odors, CO₂ content, and air distribution. A chart, completely filled in, is shown in Fig. 997.

The floor of the room is first divided into imaginary sections of approximately 200 sq. ft. each for the purpose of the measurements.

The effective temperature is determined at the center of each section, and the per cent deviation of the average effective temperature from the median line of the comfort zone corresponding to the existing air velocity is calculated and recorded. The dust particles per cubic foot of air

are then counted at the several stations. One method of counting the particles is to precipitate by impingement upon a glass disc the dust particles from a moistened and known volume of air; the particles are then counted under a microscope. The average count is plotted in column B. Culture plates are exposed for two minutes at each station, then incubated from 24 to 48 hours at 98 deg. fahr., and the number of bacteria colonies that develop are counted under a microscope; the average number is plotted in column C. The odor is determined for the room as a whole, and the test is not exact. An observer

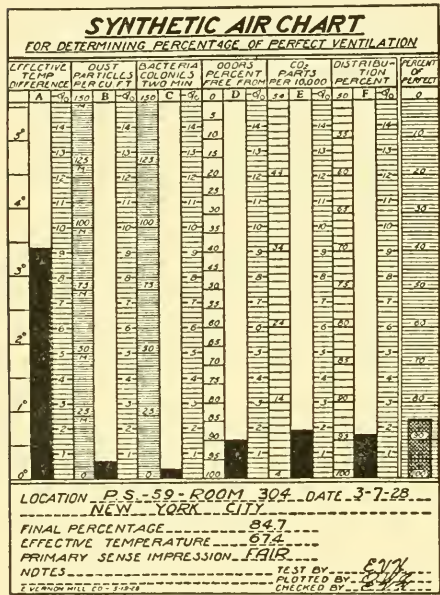


FIG. 997.—Hill Synthetic Air Chart.

goes outside of the building for a short time and then returns to the room; the test is then recorded from his observations as follows:

PRESENCE OF ODORS AND CORRESPONDING RATING

100%	95%	90%	85%	80%	75%	70%
None	Very faint	Faint	Noticeable	Distinct	Decided	Strong

The room odors may be compared with vials containing odoriferous solutions of varying strength. The CO₂ content of the air at the several stations is determined as previously described, and the average CO₂ content is plotted in column E. The CO₂ readings are also used in determining the distribution of the air. The variation in CO₂ at the various stations above or below the average content is determined, the average variation calculated, and the difference between 100 per cent and the per cent average variation is recorded in column F. Adjacent to each of the columns a penalty, in per cent, appears. The perfection

of the ventilation, expressed in per cent, is the difference between 100 per cent and the summation of the penalties; this result appears in the last column. The ventilation is usually satisfactory when the per cent perfection is above a minimum value of from 85 to 92 per cent, depending upon the use of the ventilated room.

(d) **Ventilating systems** may be classified according to the method of producing air circulation as (1) gravity or natural, (2) forced or mechanical.

(1) With **gravity circulation**, the factors responsible for a flow of air through a building are a difference between the inside and outside temperatures, the wind, and any openings through which the air may enter and leave. With no wind, if the temperature of the air inside the building exceeds the temperature of the outside air, the pressure inside is less than the outside pressure at the ground, and exceeds the outside pressure near the roof so that air will flow in at low openings and out at high levels; there would be a level called the neutral zone, at which the outside and inside pressure would exactly balance and no inflow or outflow would occur.⁵ Except in tall buildings, however, the effect of this temperature difference in producing air circulation is secondary to the pressure difference produced by wind action. When the wind blows at any angle of incidence with a wall greater than about 10 deg., a pressure is produced at the wall that is greater than would exist if there were no wind. At the same time, there is a slight vacuum produced on the leeward face of the building. If there are openings for inflow and outflow, the inside pressure will assume a value such that the quantity of air that enters is equal to the quantity of air that leaves. The direction of the prevailing wind and the orientation of the building will govern the proper location of the openings for inlet and outlet to take advantage of these regions of pressure and suction.

Ventilation by gravity circulation is widely used in residences and in single-story industrial buildings that are open throughout from floor to roof. Windows may be used to provide both the inlet and outlet of air, or they may be used as inlets, with roof ventilators as exhaust openings. Vertical flues or stacks may be installed in walls opposite windows to provide upward flow to ventilators or other exhaust openings. When the natural forces are not great, heaters may be placed in the exhaust flues to assist in circulating the air.

(2) **Mechanical ventilation** with air circulation produced by means of various types of fans is used where a positive circulation of air is desired regardless of the atmospheric conditions. Trunk and branch

⁵ See "The Neutral Zone in Ventilation," Trans. A.S.H. and V.E., Vol. 32, 1926, p. 59.

ducts carry the air under a slight pressure to air-introducing and distributing devices in the several rooms; exhaust ducts carry the vitiated air to the roof or attic. In addition to the fans that supply the fresh air, exhaust fans are used to aid in removing the vitiated air when the friction drop in pressure in the exhaust flues is high, as in multi-story office buildings. The air may be introduced near the top of the room and removed near the floor (downward distribution), or upward distribution may be used.

The upward system of ventilation has been widely used in auditoriums, theaters, and dining rooms; but the downward system, at one time used principally in class rooms, offices, and hospitals, is finding more favor for general use. Air may also be injected horizontally at a high velocity through a row of nozzles located in the upper levels of the room; a vigorous secondary circulation is then induced. In general, the air should be introduced into the room and distributed in such a manner as to avoid short circuiting and local velocities of more than 50 feet per minute in the occupied zone. For equal air velocities, the occupants of a room are more comfortable if the direction of air motion is toward their faces instead of their backs.

607. Heating Systems. General.—(a) Heating systems may be classified as to the location of the air heater with respect to the heated enclosure. A *direct* heating system is one in which the air heater, such as a radiator, is located in the heated room. With *indirect* heating, the air-heating surfaces are located outside of the room and heat is carried to the room by air that flows, by gravity or by fan action, over these heated surfaces. In any heating system, air is the ultimate carrier of the heat, while steam or hot water may act as intermediate carriers between the furnace and the air heaters.

Electric heating is limited to applications where electric energy is very cheap or where the first cost or inconvenience of any other form of heating is more important than the operating expense. The fuel burned in the furnace of any of the heating systems discussed may be coal, oil, or gas; coal is the economic choice in most localities, although the cleanliness, flexibility, convenience, and saving in space gained by the use of gas or oil in properly designed furnaces makes them very desirable fuels where their cost per unit of heating value compares favorably with that of coal.

(b) With **direct heating** by means of steam or hot water, an air heater, through which the steam or water flows, is placed within the enclosure. This air heater must have sufficient surface to supply to the room air an amount of heat that meets the heating requirements. These air heaters are commonly called direct radiators, although the name is a

misnomer, since heat is transferred from the radiator surface to the room air principally by convection. Radiators are commonly made of cast-iron sections joined together by means of nipples; in industrial buildings, radiators consisting of pipe coils are often used.

The amount of heat transferred from the radiator to the room air by radiation and convection per unit of time depends upon the area of the radiator surface, the mean temperature of that surface, the temperature of the room, the nature of the radiator surface, the height and shape of the radiator, and its location in the room. Cast-iron radiators are rated by the manufacturers on the basis of a heat emission of 240 B.t.u. per hour per rated square foot of surface with a steam or water temperature of 215 deg. fahr. and a room temperature of 70 deg. fahr. The rated surface of column, tubular, wall, or window radiators of different heights and number of sections may be obtained from the manufacturers.

Radiators are located, preferably, beneath the windows in a room. With this location, the infiltrating air is warmed, passes upward, across the ceiling, and down the opposite wall before it comes into contact with the occupants of the room. If the radiators were placed along an inside wall, the cold air from the windows would cross the room near the floor, before being warmed, and would chill the occupants.

Each type of direct heating system will be discussed separately in the sections immediately following; and indirect heating systems will be discussed in Sect. 610.

608. Hot-Water Heating Systems.—(a) In a hot-water heating system, water is the intermediate carrier of the heat. Circulating in a closed system, the water absorbs heat from the products of combustion in the furnace and gives up heat through the radiators to the room air. In gravity circulation systems, the force that circulates the water is caused by the difference between the weight of the hot water in the supply risers and the weight of the cooler water in the return lines.

(b) The principle of water circulation in a **gravity hot-water heating system** may be explained with reference to Fig. 998. In (a), there is no flow occurring, and the water in the left column is at the temperature

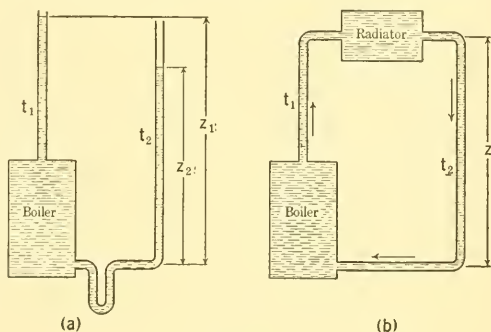


FIG. 998.

t_1 and has a density of ρ_1 lb. per cu. ft., while the water in the right column is at the temperature t_2 (less than t_1), and has a density of ρ_2 lb. per cu. ft. For equilibrium, if the cross-sectional area of each column be A sq. ft.,

$$\rho_1 AZ_1 = \rho_2 AZ_2. \quad . \quad . \quad . \quad . \quad . \quad . \quad (684)$$

Since $t_1 > t_2$, $\rho_1 < \rho_2$, and $Z_1 > Z_2$.

If the two pipes are now connected through a radiator, as shown in (b), the equilibrium is destroyed, because the height of both the columns of water is now the same, and a flow will start in the direction shown. The force, in pounds, that produces this circulation is caused by the difference in density of the water in the two columns of equal height and is given by the equation,

$$F_c = Z(\rho_2 - \rho_1)A, \quad . \quad . \quad . \quad . \quad . \quad . \quad (685)$$

in which Z = the vertical distance between the outlet of the water from the radiator and the inlet to the water heater, in ft.

If the pressure head available for circulation, in feet of water, be denoted by Z_c , then

$$Z_c A \rho_m = Z(\rho_2 - \rho_1)A$$

or

$$Z_c = \frac{Z(\rho_2 - \rho_1)}{\rho_m}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (686)$$

in which ρ_m = the mean density of the water in the system, in lb. per cu. ft., commonly taken at the mean temperature, $\frac{1}{2}(t_1 + t_2)$.

The variation in the density of water with temperature is shown in Fig. 999.

A fundamental principle used in the selection of the proper size of pipe for any heating system is that a flow of the circulating medium is *set up* when the head tending to produce circulation balances the sum of the velocity head and all losses of head. A steady flow of the circulating medium will be *maintained* when the head tending to maintain circulation balances all losses of head. The loss of head due to friction in pipe and fittings may be estimated by the methods explained in Chapter XLII.⁶

⁶ Also see Trans. A.S.H. and V.E.: Vol. 21, 1915, p. 473; Vol. 23, 1917, p. 499; Vol. 35, 1929, p. 419; also Heating, Piping, and Air Conditioning: Aug., 1929, p. 267, and July, 1933, p. 381.

(c) In hot-water heating systems using **forced circulation**, the circulating head may be supplied by a small centrifugal pump inserted in the return main, near the entrance to the water heater. Smaller pipes and radiators than are necessary with gravity circulation may be used. The resistances in the several water circuits must be balanced against the respective circulating heads exactly as in designing a gravity circulation system.

(d) Hot-water heating systems are widely used in residences where the good overall efficiency of the system and the ease of regulating the water temperature to meet the heating requirements are definite advantages. Hot water may also be circulated by pumps from a central plant to buildings equipped with the necessary radiator and service piping.

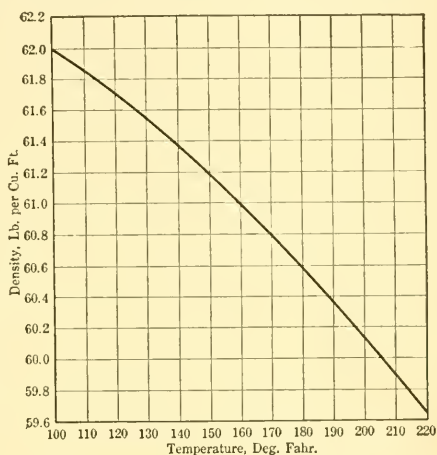


FIG. 999.—Density of Water.

A central plant is usually equipped with steam boilers furnishing the steam required to drive the circulating pumps; the exhaust steam is condensed in water heaters, and water-heating boilers are commonly installed in series with these exhaust heaters. Since the temperature of the water in the radiators is usually less than when

steam is used as the heating medium, hot-water radiators commonly require a greater heating surface than steam radiators.

609. Steam Heating Systems.—(a) In a steam heating system, steam is the intermediate carrier of heat between the furnace and the room air. Steam is generated in a boiler, flows

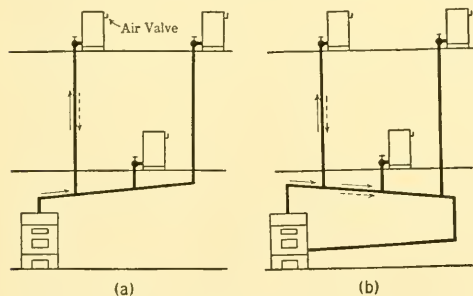


FIG. 1000.—Air-Vent Single-Pipe Systems.

to the air heaters (radiators or pipe coils), condenses within these air heaters, giving up heat through them to the air, and the condensate is generally returned to the boiler. Steam heating systems may be roughly classified according to the steam pressures and according

to the methods used in eliminating air from the system and in returning the condensate to the boiler. Classified in this way there are four different systems: (1) the air-vent single-pipe, (2) the air-vent double-pipe, (3) vapor, and (4) vacuum.

(b) **Simple air-vent single-pipe systems** are shown diagrammatically in Fig. 1000 (a) and (b), with piping details omitted. In both plans shown, the steam pressure is above atmospheric and air may be vented from air valves located on the radiators; and only one pipe carries steam to, and condensate from, each radiator, as shown. In the arrangement shown in (a) the main pitches toward the boiler, and there is a counter flow of live steam and condensate in the main as well as in the risers. Unless the velocity of the steam in the main is low, the condensate may be picked up by the steam and carried to the end of the main with consequent "water hammer." In the arrangement shown in (b), the main pitches away from the boiler, and although there is a counterflow

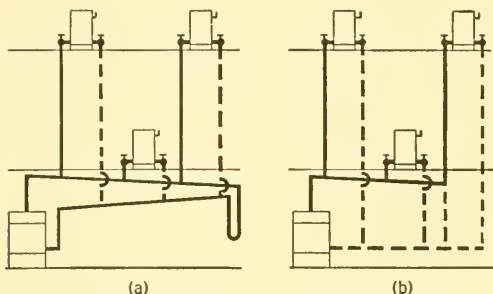


FIG. 1001.—Air-Vent Two-Pipe Systems.

of steam and condensate in the risers, the two fluids flow in the same direction in the main. Counterflow may also be partly eliminated by an overhead distribution system.

(c) **Air-vent two-pipe systems** are shown diagrammatically in Fig. 1001 (a) and (b). With steam pressures above atmospheric the air may be vented as in the one-pipe systems. Each radiator is, however, provided with both a supply and a return pipe so that there is a passage of steam through the radiator and into the return line. In (a), a *dry return system* is shown. The return line is above the water level in the boiler, and carries steam as well as condensate. A water seal is provided at the end of each supply main in order to prevent the pressure in the return line reaching the supply pressure. A *wet return system* is shown in (b). The return main is below the water level in the boiler and carries only condensate. Each radiator and the low points in the supply line are connected to the return line. Since the condensate and the steam do not flow together in any horizontal pipes, the wet return system will ordinarily operate with less noise than the dry return.

In any of the preceding systems, air lines may convey the air through a thermostatic type air valve from all the radiators to a vacuum pump

or ejector by means of which the air may be positively eliminated without steam or water leakage.

(d) **Vapor heating systems** are a modern and popular form of the two-pipe system in which low pressure steam is used and in which the radiators pass only air and condensate into the return line. There are several commercial types of vapor systems differing principally in the types of inlet valves and traps used in the radiators and in the methods of venting the air and of returning the condensate to the boiler. The fundamental arrangement of the vapor system is shown diagrammatically in Fig. 1002.

The inlet valves are designed to control the amount of steam admitted to the radiators; and the traps, either by means of a thermostatic or float arrangement, permit only air and water to enter the return line. With the air previously driven from the system, banking the fires in the furnace of a vapor system will cause the pressure of the steam to drop below atmospheric, and vacuums of as much as 15 in. Hg are carried. When the combustion rate is high, the steam pressure slightly exceeds atmospheric. The elementary system shown in Fig. 1002 might give trouble under high steam pressures, due to a flooding of the return line and a failure of water to enter the boiler, since sufficient static head must be built up in the return line to balance the boiler pressure before any water will enter. To prevent flooding the return line, a water return without this high static head may be secured with a boiler

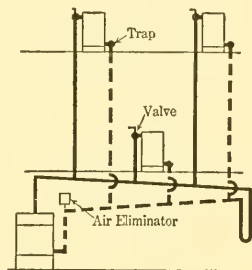


FIG. 1002.—Elementary Vapor System.

feed pump, an alternating return trap, or a differential seal. A boiler feed pump is frequently used in large systems and the alternating trap or differential seal in smaller units. The alternating trap connects both to the return and supply mains, and is vented through an air relief valve. As the boiler pressure builds up, the water level rises in the trap to a point at which a float arrangement closes the air vent and opens a valve connecting to the supply main; with boiler pressure over the water surface, a small static head is sufficient to force the water from the trap into the boiler. The differential seal is shown in Fig. 1003. The water in the left leg, *A*, of the loop is exposed to the pressure of the steam in the supply line, while the water in the right leg, *B*, is open to the pressure in the return line. The position of the water shown is for a small difference between these pressures. When the differential pressure becomes sufficiently great, the water in *A* will be pushed down until the lower end of the pipe *C* is opened, and thus the pressure in the

supply line will be transmitted to the return line. Water will then be returned to the boiler regardless of the boiler pressure.

(e) The **vacuum heating system** is quite similar to the vapor system, except that *vacuum pumps* maintain a pressure below atmospheric in the return lines. The temperature of the steam in the radiators can be controlled through the vacuum supplied by the pump, and the range of temperatures secured in a hot-water heating system can be approached in a good vacuum system. Since the flow in the return lines does not depend upon the action of gravity, radiators in a vacuum system can be located, if desired, below the water level of the water in the boiler.

(f) The factors that must be considered in the rational determination of the **proper size of pipe** for a steam heating system are the heating requirements at the several radiators or other air heaters, the available

pressure head for producing circulation and the friction losses in pressure head in the pipes and fittings, the effect of the velocity of steam upon counterflow of the steam and condensate, the cost of the piping, and the cost of the insulation required to keep the heat losses from the pipe to an economic minimum. The determination of the steam flow rate necessary to meet the heating requirements is based on the assumption that the quantity of heat given up by each pound of steam to the room air is the latent heat of condensation corresponding to the steam pressure. Instead of a rational determination of the most economical size of steam pipe, the empirical method, based upon previously satisfactory practice, of allowing a pressure drop of 1 ounce per

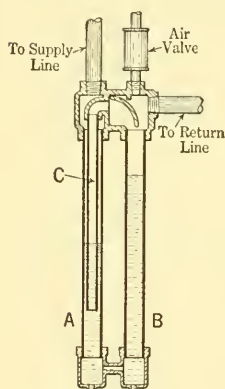


FIG. 1003.—Differential Seal.

sq. in. per 100 ft. of run is often used in fixing the size of pipes in which steam and condensate flow in the same direction. The pressure losses up to all points of delivery may then be roughly equalized in order to prevent short circuits in the flow.

The drop in pressure due to friction in steam heating lines may be found, using Fig. 847 and Reynolds' number, as explained in Sect. 505, p. 816, or by using the Unwin-Babcock empirical formula, given in Eq. (616), p. 822. The Unwin-Babcock formula holds very closely for the state of the steam at low pressures as it is used in heating systems, and graphical solutions of this formula, such as the one in Fig. 1004, are very useful in selecting the pipe sizes for heating installations. For example, with a steam flow rate of 2000 cu. ft. per hr. in a pipe of 2 in. nominal diameter, from Fig. 1004, the velocity of the steam is about

23 ft. per sec., and the friction drop in pressure is about 1.5 ft. of steam per foot of pipe or, assuming a reasonable average density of the steam of 0.04 lb. per cu. ft., the pressure drop is 0.7 oz. per sq. in. per 100 ft. of pipe.

Where the flow of condensate is counter to the steam flow, a pipe size is selected that prevents interference in the flow of these two fluids. In upflow risers in a single-pipe system, for example, so long as the velocity of the steam is low, the condensate flows down the inner wall of the pipe, the steam up the middle, and there is little interference. If the pipe is too small for the required flow rate, the velocity of the steam may be sufficient to cause the water to be picked up at the bottom of the riser and carried into the radiator with an accompanying cracking noise. The counterflow of steam and condensate has received much experimental attention, and the proper pipe sizes in the flow have been determined.⁷

610. Indirect Heating Systems.—(a)

The indirect heating system with gravity circulation of the warm air consists of a furnace, warm-air pipes and registers, and cold-air registers and pipes. The cold-air supply to the furnace may be taken (1) entirely from inside of the

building through cold-air registers and cold-air pipes, (2) entirely from outside the building with no recirculation, or (3) partly from within and partly from outside the building. This air circulates between the furnace casing and surfaces that are heated by contact, on the other side, with the hot products of combustion. In the piped installation, the warm air flows through horizontal pipes called leaders into vertical pipes called stacks or risers that discharge into registers in the several rooms. A pipeless system is sometimes used in small residences, in which there is but one register located in a room immediately above the furnace. The communicating rooms are then warmed by the natural circulation of the air within the dwelling.

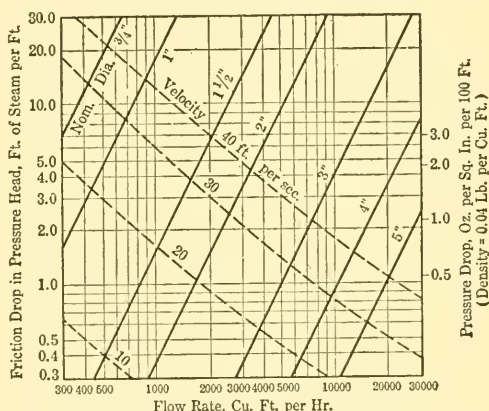


FIG. 1004.—Graphical Solution of the Unwin-Babcock Formula.

⁷ Trans. A.S.H. and V.E.: 1923, p. 109; 1924, p. 323; 1925, p. 135; 1927, p. 289; 1929, p. 399.

With the piped system, the quantity of heat that can be supplied to a room depends upon the temperature and the weight of air discharged into that room. The weight of the air discharged into a room in unit time depends, in turn, upon the circulating head, i.e., the difference between the densities of the warm and cold air and the stack heights, the cross-sectional area of the air pipe, and also upon the frictional resistance to air flow offered by the air pipes. Since the circulating head is small, it is important that the air pipes be of sufficient size and so constructed that there is very little turbulence and resistance to air flow. Furthermore, the air pipes should be insulated thermally, and the stacks should be located in inside partitions to prevent chilling.

In designing a gravity-circulation warm-air installation, the first

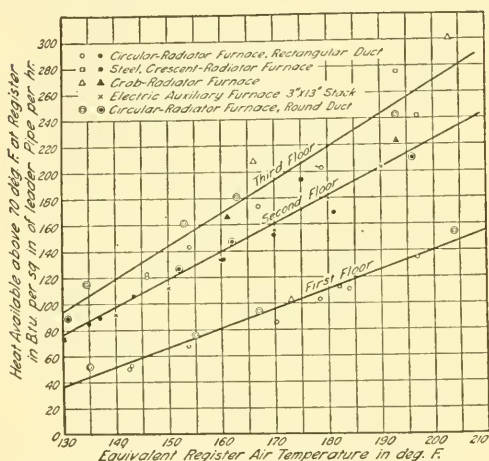


FIG. 1005.—Results of Tests on Gravity Warm-Air Heating Systems.

step is the selection of a furnace of sufficient grate and heating surface area to meet the heating requirements of the entire dwelling. The leaders, stacks, and registers for the several rooms are then so proportioned that there will be a sufficient flow of warm air to balance the heat losses with the existing differences between the respective circulating heads and the frictional resistances. The results of tests conducted at the University of Illinois⁸ on several simple gravity warm-air installations, having straight leader pipes 8 feet long are shown in Fig. 1005. From data like this, the cross-sectional area of the leaders may be estimated for any given register temperature to meet the heating requirements.

(b) *Fans* may be used in a warm-air heating installation, either to assist a circulation of air that is primarily dependent upon density differences, or to provide a circulation that depends largely on the pressures produced by the fan. In the latter case, the warm air is usually distributed through a trunk system of ducts and branches instead of through individual leaders and stacks, the fan is usually of the centrifugal type, and lower register temperatures are used because of the

⁸ From University of Illinois Eng. Expt. Sta. Bull. No. 141, 1924, p. 121.

greater weight of air circulated. In either case, the fan is usually arranged to blow the cold air through the furnace casing. With a circulation of air that is substantially independent of density differences, that duct through which there is the least frictional resistance to air flow will transmit the greatest weight of air in unit time regardless of the stack height.

(c) Gravity-circulation warm-air heating systems are widely used in residences, stores, and other small buildings where the low first cost of the installation is deemed very important. In small sizes, the efficiency of the warm-air system using recirculation should be as high as that for hot water and higher than for steam heating. The system is extremely flexible, and some ventilation can be accomplished by the circulation of the air. Since there is no danger of freezing, such systems are commonly used in buildings that are heated intermittently. When the circulation of the air is forced by a fan, larger buildings, such as schools, can be heated with warm-air systems. The furnace casing and all air pipes must be air tight, otherwise coal gas, smoke, ashes, and dust may be circulated with the air; also, the entire system must be very carefully designed and built to secure an even distribution of heat.

(d) Indirect heating with **steam** or **hot water** as the intermediate carrier of heat is used in large buildings, such as factories, where direct heating is not economical or not desirable, and where the functions of heating and ventilation must be combined. If the heating requirement is entirely met by air circulated by fans through heaters and ducts to the enclosure, the system is called a *central fan or hot-blast system*. The *split system*, in common use in office buildings and schools, is a combined direct and indirect system of heating; the heat required to balance that lost through the building and to warm infiltrating air is supplied by direct radiators, while the fans circulate the air necessary for ventilation through the air heaters and thence through ducts to the rooms.

A common arrangement of a fan system of indirect heating in a factory is shown in plan in Fig. 1006. Either steam or hot water may be circulated through air heaters that consist either of pipe coils, finned tubes, or cast-iron sections connected by nipples into stacks or units. Since the amount of hot air necessary to meet the heating requirements commonly exceeds that necessary for ventilation, heating economy may be secured by recirculating some of the heated air. The damper shown on the suction side of the heater controls the relative amounts of cold and recirculated air admitted. The distributing ducts are made of galvanized steel or iron or formed in masonry with smooth interiors, and are covered with insulating material. The arrangement of the

ducts in Fig. 1006 is known as a **trunk-line** system, in which the fan discharges into a main duct with individual branches taken off at intervals. In another type of indirect heating system, common to schools, air heated to about 70 deg. fahr. is discharged by the fans into a large **plenum chamber** located in the basement under a corridor. From the plenum chamber, the air is distributed by ducts to the several rooms. An air heater is installed in the plenum chamber for each tier of rooms, and a mixing damper under thermostatic control supplies the desired mixture of heated air and tempered air.

(e) The **rate of heat transmission** from the steam to the air in the air heaters used in indirect heating systems depends upon the heating surface, the logarithmic mean temperature difference between the condensing steam and air, and the overall coefficient of heat transmission of the heater. This coefficient, in turn, depends primarily upon the type of heater and upon the air velocity through the free area of the

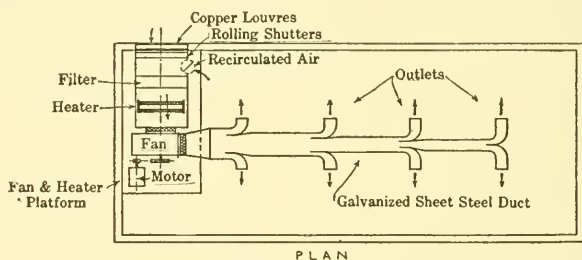


FIG. 1006.—Fan System of Heating.

heater; values of this coefficient may be secured from the manufacturers of the heaters.

The friction drop in pressure of the air in passing through the heater increases nearly with the square of the air velocity, and tables are furnished by the manufacturers giving values of this pressure drop. The selection of the heater is a problem that involves not only the cost of the air heater but also that of the fan. For example, a heater of small surface and low first cost may give the desired rate of heat transmission at high air velocities. At the same time the pressure drop through the heater may be so great that a large fan of high first cost and high operating costs may be necessary to provide the required air circulation.

(f) The **design of the air ducts** is a similar problem, for although the pressure drop in the air ducts, and therefore the fan costs, may be kept low by using ducts of large cross-section, the cost of these ducts and their insulation will increase with their size. As a first approximation to the correct economic balance, the assumption is frequently made that the pressure drop in the heater should be from 40 to 60 per cent

of the entire pressure loss in the air line. Then, in trunk-line distribution systems, the ducts are proportioned to give equal friction losses in pressure per foot of length. In the plenum system, the duct having the largest pressure drop is designed first; the pressure that must be carried in the plenum chamber is then fixed, and the other ducts are then designed to deliver the required volume of air with the known pressure drop.

The friction drop in pressure may be estimated by the methods explained in Chapter XLII. The value of Reynolds' number for air at 120 deg. fahr. and at a pressure near atmospheric, flowing with a velocity of 1000 ft. per min. through a round duct, 20 in. in diameter, would be about 1.5×10^5 . Hence, from Fig. 847, for a galvanized iron or steel duct, the friction factor, f , would be nearly 0.02. A value of 0.024 is frequently used for f in proportioning duct systems, and this value is apparently well on the safe side; this value of f was used in preparing Fig. 1007, in which the friction drop in pressure of air flowing in round ducts may be estimated. The friction drop in pressure is given in the units of feet of air; this may readily be changed to pounds per square inch, when the pressure and temperature of the air are known.

Assuming the friction factor, f , to remain constant and equal to 0.024, the loss of head due to friction in rectangular ducts may also be found from Fig. 1007. For rectangular ducts with sides a and b inches, the *equivalent diameter* of a round duct to give the same rate of flow, in cubic feet per minute, and the same friction drop in pressure, may be shown to be

$$d_e = 1.266 \frac{(ab)^{0.6}}{(a + b)^{0.2}} \quad \dots \quad (687)$$

Example: Find the friction drop in pressure in a rectangular duct, 12 in. by 24 in., 50 ft. long, carrying 2000 cu. ft. of air per min. From Eq. (687), the equivalent

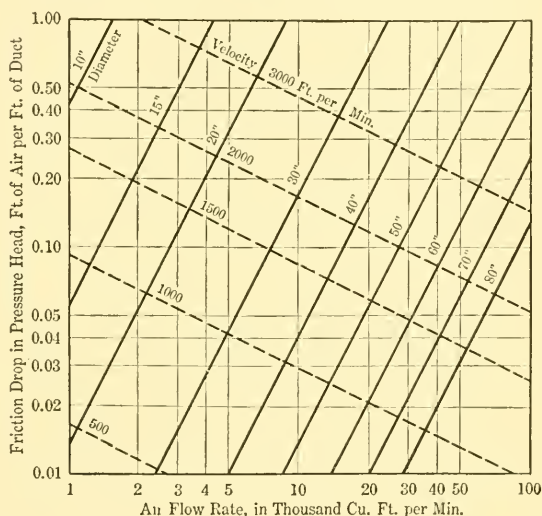


FIG. 1007.—Friction Drop in Pressure for Air in Round Ducts of Fan Systems.

diameter of the round duct is 18.5 in., and from Fig. 1007, for this diameter and a rate of flow of 2000 cu. ft. per min., the friction drop in pressure is about 0.08 ft. of air per ft. of duct, or about 4.0 ft. of air for this duct.

611. Central and District Heating.—(a) Either steam or hot water may be distributed from a central plant to several buildings. A **central heating** system is one that serves a group of buildings belonging to a single institution, such as a university or a hospital, while the term “**district heating**” usually refers to the heating from a central plant of public buildings, stores, offices, or residences in a limited district of a city. A central or district heating system may require less labor, be cheaper to install, and be more efficient in fuel consumption than individual heating plants for the several buildings. The loss of heat and drop in pressure of the heating fluid in the distributing mains and the cost of such mains limit the radius that can be economically served, but a distributing radius of one mile or more is not uncommon. Although hot water is sometimes used as the heating medium in central or institutional heating, steam is used in most district heating systems. The disadvantages in the use of hot water for district heating are difficulties encountered in metering, the high water pressure and heavy piping required, the unsuitability of hot water at moderate temperatures for cooking and laundry use, and the high cost of the distributing system.

The heating requirements and the selection of the service piping and radiators for each consumer involve the same principles that have previously been discussed, but the distribution and metering of the steam in district heating are new problems.

(b) Any system of **steam distribution** by underground piping must be designed to protect the pipe from mechanical injury and corrosion, to drain the condensate, to provide for expansion and contraction, and to prevent excessive heat loss. The pipes may be placed in tunnels or in protective conduits made of wood, tile, concrete, or brick. Corrosion may cause rapid deterioration and loss of carrying capacity of the distributing pipes and, particularly, the return lines. The greatest corrosive action is on the inside of the pipe due, primarily, to active acidity and the presence of oxygen.⁹ Chemical treatment and deaeration of the boiler feed water are measures that may be taken to prevent this corrosion. Ground water that seeps into tunnels or conduits must be drained to prevent corrosion of the outside pipe surfaces. In long pipes expansion joints are provided at intervals of from 60 to 300 ft., depending upon the nature of the joint. The pipes are anchored midway between

⁹ See “Combating Corrosion of Piping,” by Joos and Rohlin, *Heating, Piping, and Air Conditioning*, 1932. August, September, and October.

the expansion fittings, and branch pipes are connected near these anchors. Underground pipes are pitched about 1 in. in 50 ft., and the condensate is drained through traps from all low points and at intervals of from 200 to 300 ft. Due to the insulating effect of the soil, the pipes are usually buried at least 2 ft. under the surface of the ground; the thermal conductivity of the soil depends upon its nature and moisture content, but is about 2 or 3 B.t.u. per hr. per deg. fahr. per sq. ft. per in. for dry soil and approximately five times as great for wet soil.

(c) The **meters** used to measure the weight of steam used by the consumers may be of either the **condensation** or **steam flow** type. The condensation meter is installed at the discharge end of the return piping, and although this is a simple and convenient method, any steam or condensate that leaks from the consumer's piping is not metered. The steam can be metered at entrance to the consumer's piping by steam flow meters of the displacement, orifice, or venturi type with mechanical or electrical transmitting and recording devices.¹⁰

612. Unit or Room Coolers.—(a) Unit or room coolers are used when the air in an enclosure is to be cooled with no provision for heating or year round air conditioning. These coolers may be of the cabinet or portable type with self-contained or remote refrigeration. The cooling medium may be ice, cold water, or a refrigerant.

(b) In a typical portable cooler using ice as the cooling medium, small multi-blade centrifugal fans draw the room air across the ice and then discharge it through adjustable shutters into the room. The cabinet is made of steel, insulated, and mounted on rubber-tired swivel casters. With a charge of 300 lb. of ice, one of these coolers has an average cooling capacity of 9600 B.t.u. per hr. for about 5 hours, when the entering dry-bulb temperature is 80 deg. fahr., the entering relative humidity is 50 per cent, and the fans circulate 400 cu. ft. of air per min. A tank, located below the grid holding the ice, is large enough to hold the 300 lb. of water and any moisture condensed from the air. This tank is provided with a drain. Since these coolers only require attachment to an electric outlet and drain, they may be used in existing rooms where no provision has been made for any other form of cooling.

(c) Recently, unit room coolers with **self-contained refrigeration** have been introduced. These coolers are equipped with small compressors (driven by motors of $\frac{1}{3}$ to 1 hp.), which deliver the refrigerant through cooling coils, over which the air is circulated by fans. Either freon or methyl chloride is used as the refrigerant in these units, which

¹⁰ For a complete discussion of district heating, see the current Handbook of the National District Heating Association.

compete with the ice-cooled ones. (Refrigerants will be discussed in the next chapter.)

Some of the room coolers are equipped with heating coils, through which steam or hot water may be circulated; and these units are usually provided with air filters, air humidifiers, and automatic or manual control of temperature and relative humidity. In such cases, the unit gives year round conditioning of the air, and is properly called a *unit air conditioner*.¹¹

613. Air Conditioning.—(a) Air conditioning may be practiced for the purpose of promoting human health and comfort, improving laboring conditions and human efficiency, maintaining material in the most suitable state while undergoing manufacturing operations or while in storage, or supplying air suitable for drying or other industrial processes. In a strict sense, unless year round control of at least temperature, humidity, and air motion are provided, no equipment may properly be designated as “air conditioning.”

Air conditioning for the purpose of promoting human comfort must control effective temperature if effective temperature is a true index of human comfort. The type of equipment used to control effective temperature depends upon the size of the installation. Two types of equipment will be described, one for use in large buildings such as office buildings, stores, and theaters, another for use in residences.

(b) **Air-conditioning equipment** has evolved from the *air washer* which has been used in large buildings for many years. In the air washer, air is forced through a chamber containing a water spray and then over baffle plates which eliminate free moisture and wet particles of solid matter. Although the air washer was initially used for the removal of dust from the air, its use will always cause an increase in the water vapor content and a decrease in the dry-bulb temperature of the air, if the same spray water (with makeup) is used repeatedly without any external heat transfer to or from the air while passing through the washer. Under these conditions, the process is one of adiabatic saturation at substantially constant wet-bulb temperature. By the addition of heating coils to the washer, the temperature of the air may be raised with simultaneous humidifying, as before, by the water spray. The air washer may also be operated to dehumidify the air and lower its temperature by supplying spray water at a temperature below the dew-point of the entering air; in this case, the spray water may be cooled by the use of mechanical refrigeration, steam jet refrigeration, or ice. The relation of the outside temperature and humidity of the air to the effective temperature that is desired in the conditioned enclosure deter-

¹¹ See Heating and Ventilating, July, 1933, p. 10.

mines which of these operations shall take place in the air-conditioning equipment.

(c) *Humidification* of the air is necessary to prevent an excessively low relative humidity of the inside air during the winter months. If outside air at a dry-bulb temperature of 0 deg. fahr. and a relative humidity of 100 per cent be heated to 70 deg. fahr., without the addition of water vapor, the final relative humidity is only about 5 per cent. The amount of water vapor which can be added to the air is limited, however, by the temperature of the inside surface of the windows and walls of the enclosure.¹² For the above outside and inside temperatures, the temperature of the inside surface of a double glass window might be only 50 deg. fahr. The dew-point of the inside air could not be higher than this without condensation of water vapor on the windows; the indoor relative humidity would then be limited to about 50 per cent for double windows, and the addition of about 0.007 lb. of water vapor to each pound of dry air would be necessary to produce this relative humidity. In the modified spray washer, freezing of the spray water is prevented by heating the air in two stages; the air is first heated to about 40 deg. fahr. by passing over tempering coils, next sprayed with heated spray water and then heated to the required temperature by the main heating coils.

(d) **Cooling** of the air by *humidification*, or *cooling with dehumidification*, may often be necessary to provide the proper effective temperature for human comfort during the summer months. Referring to Figs. 969 and 994, if the outside dry-bulb temperature is 85 deg. fahr. and the wet-bulb temperature 70 deg. fahr., the dry-bulb temperature can be reduced, at constant wet-bulb temperature by *humidifying* the air with uncooled, recirculated spray water, to a dry-bulb temperature of 70 deg. fahr. as a limit; if the dry-bulb temperature is actually reduced to 75 deg. fahr., the effective temperature will be lowered from 77.5 to 72.5 with an *addition* of about 0.0024 lb. of water vapor per lb. of dry air. This illustrates the cooling that can be secured by *humidification* without artificial cooling of the spray water.

By *dehumidifying* and *cooling* the air, the same final effective temperature can be obtained with a smaller reduction in dry-bulb temperature. With the same outdoor conditions as in the preceding illustration, the effective temperature could be reduced to 72.5 with a final dry-bulb temperature of 80 deg. fahr. (instead of 75), by condensing about 0.0043 lb. of water vapor per pound of dry air. In this case, the air would be "saturated" in the modified air washer at 51 deg. fahr. (the dew-point corresponding to the desired indoor conditions), and intro-

¹² See Trans. A.S.H. and V.E.: 1923, p. 49; 1930, p. 447.

duced into the room, where liberation of heat or controlled mixing with unconditioned air would raise the dry-bulb temperature to 80 deg. fahr. If a rise in temperature of the spray water of 10 deg. were allowed, this water would be supplied at 41 deg. fahr. and would need to be artificially cooled. The reduction of effective temperature by dehumidifying is favored over air cooling by humidifying where the reduction in dry-bulb temperature, necessary with the latter method, is excessive.

(e) **Methods of dehumidifying the air**, industrially, other than by spraying with water at a temperature below the dew-point, are chemical adsorption of the vapor from the air by gels, or absorption by calcium chloride, or concentrated sulphuric acid, or surface condensation of water vapor produced when the air is passed over surfaces that are artificially cooled. Silica gel, a colloidal form of silicon dioxide, has been used to maintain a relative humidity of 0.5 per cent in telephone cable storage ovens.¹³ The silica gel adsorbs water vapor from the air at 68 deg. fahr.; and later in the cycle the gel is activated by passing air at 400 deg. fahr. through the gel to carry off the moisture.

(f) A **typical installation** for humidifying and heating air is shown in plan view ¹⁴ in Fig. 1008. Air from the outside, from the enclosure being conditioned, or from both sources enters the apparatus through controlled dampers under fan suction. The air then passes through the humidifying chamber, the water-eliminating element, and the air heater (or by-pass), to the fan which delivers it into the distribution ducts. The water is pumped through spray nozzles, collects in the settling chamber at the bottom, passes through strainers for the removal of dirt, and returns through a heater to the pump. This installation is only one of many possible arrangements; with the removal of the heaters, and the substitution of cooling coils through which a refrigerant or cold water could be circulated, or by the installation of refrigeration to cool the spray water, the apparatus could be changed to operate as an air cooler and dehumidifier.

(g) Special **control instruments** have been developed to govern automatically the regulating valves and dampers of air-conditioning apparatus. The pneumatic system of control is used in the apparatus shown in Fig. 1008, but electric systems are also used. A *dew-point thermostat* located in the path of the "saturated" air issuing from the humidifying element may control the temperature of the spray water or the entering air, or both, or the proportions of fresh and recirculated air admitted in order to fix the temperature at which the air is "saturated" (the dew-point); a second thermostat placed in the conditioned enclosure may

¹³ Power, March 22, 1932, p. 429.

¹⁴ From Heating, Piping, and Air Conditioning, Nov., 1929, p. 608.

regulate the dry-bulb temperature, and consequently the relative humidity, in the enclosure by controlling heating elements at the humidifier outlet or by varying the volume of conditioned air admitted to the room.

When a constant relative humidity is to be maintained in a conditioned enclosure regardless of variations in the dew-point or dry-bulb temperature, a *hygrostat*, containing a hygroscopic element which responds to changes in relative humidity, can be applied to control the volume of "saturated" air admitted to the room from the humidifier or to vary the temperature or quantity of the spray water.

For control of dehumidifying and cooling, the dew-point thermostat may control the temperature of the spray water coming from the refrigerating equipment, and the room thermostat may control the volume of dehumidified air supplied to the conditioned enclosure.

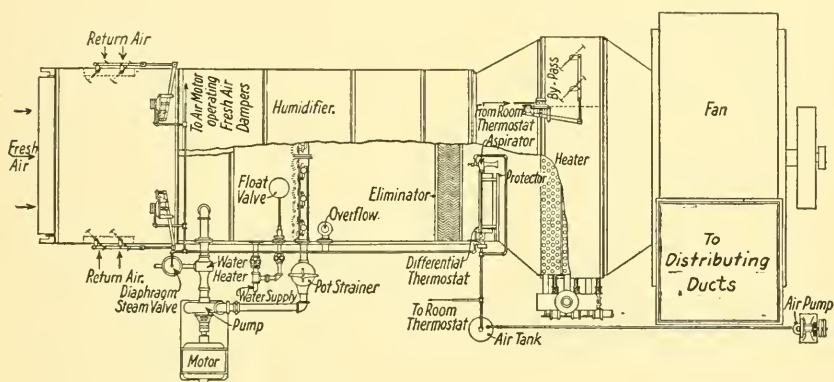


FIG. 1008.—Air Heater, Humidifier, and Cleaner.

(h) Air-conditioning equipment that *cools* and *dehumidifies* the air during hot weather may also be used to *warm* the air during cold weather by using the refrigerating equipment as a warming machine. As applied to the heating of an office building in Los Angeles,¹⁵ the water that absorbs heat from the condensing refrigerant (methyl chloride) is pumped through heaters located in the path of the air entering the building. The liquid refrigerant passes through the expansion valve and is vaporized by absorbing heat from water circulated through the cooler; this circulating water is then heated by its passage through an air washer, on the building roof, within which the water absorbs heat from the outside air. Such operation is economical only when electrical energy is cheap and when the temperature difference to be maintained between the inside and outside air is small. In the application men-

¹⁵ Power, July, 1932, p. 29.

tioned, the building was equipped for direct heating by electricity, and since the overall coefficient of performance of the heat pump was between about 1.5 and 2.0, this is the ratio of the energy cost of heating directly by electricity to the energy cost of heating by the heat pump. The fixed costs of all extra equipment must also, of course, be charged against the heat pump in a complete study of the economy of this method of heating.

(i) **Residential air-conditioning equipment** of a "central" type in addition to the unit conditioners, previously mentioned, is now available. Most of the residential air-conditioning equipment intended for installation in new residences provides fan circulation of air over filtering, humidifying, heating, and, if desired, cooling and dehumidifying ele-

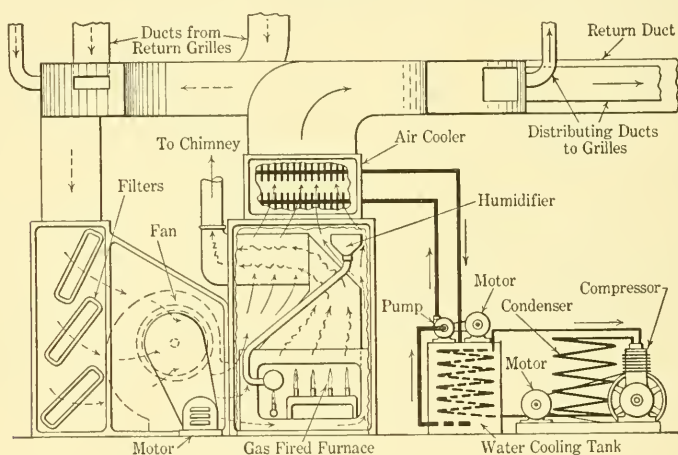


FIG. 1009.—Residential Air Conditioning.

ments, and distribution of air through ducts and grilles to the several rooms with automatic control of temperature and humidity. The heating of the air may be effected by the use of steam or hot-water coils or by heat exchangers through which the products of combustion pass, as in a warm-air heating installation. The heating capacity of most of the equipment on the market at the present time is within limits of 80,000 to 300,000 B.t.u. per hr. Humidifying is accomplished by spraying the air with water or by evaporation of water in pan-type humidifiers placed in the air stream and heated by the products of combustion. Cooling is effected by spraying the air with cold water or by forcing the air over cooling elements of the surface type through which the cooling medium flows; a refrigerant, such as freon or methyl chloride, or water, cooled by ice or by mechanical or steam jet refrigeration, may

be used as the cooling medium. In the last case, the boiler that supplies the steam or hot water for winter heating may also furnish the steam required for summer cooling. Mechanical refrigerating units are available for residential use with cooling capacities of from about 6000 to 60,000 B.t.u. per hr.

(j) A **typical installation** of air-conditioning equipment in a residence is represented, diagrammatically, in Fig. 1009. The heating system is of the forced warm-air type, with fan circulation of air through a heat exchanger warmed by the products of combustion from a gas-fired furnace. Air filters are placed in the return air line, and a humidifier of the pan type is installed within the furnace casing. A mechanical refrigerating unit is installed for summer use, and the cooling element is of the surface type, through which water, cooled by the refrigerant, is circulated. The distributing and return ducts are shown. No attempt has been made in the figure to show all of the connections, piping, and wiring required.

(k) Air conditioning is widely used in **industry**, but the requirements of different industrial processes are so widely different that each problem is individual. The relative humidity of the air is important in the manufacture and processing of all hygroscopic materials such as textiles. The satisfactory operation of many automatic machines, such as those used for wrapping chewing gum, confectionery, food products, and cigarettes, requires close control of the temperature and humidity. Air conditioning is also employed in the rubber industry, the printing and lithographing industries, the processing and weaving of artificial silk, and the manufacture of food products, confections, cigars, and photographic film.¹⁶

614. Maximum Heating and Cooling Loads.—(a) The maximum heating and cooling loads must be estimated before the air-conditioning equipment for a given installation can be selected. The principal factors that determine the maximum heating load are (1) the minimum outside dry-bulb temperature and corresponding absolute humidity, (2) the desired indoor temperature and absolute humidity, (3) the building construction and arrangement, (4) the amount of outside air entering the enclosure by infiltration as determined by (3) and the velocity of the prevailing winds, (5) the amount of outside air circulated for ventilation, and (6) the amount of heat and water vapor given off within the enclosure by persons, lights, machinery, and processes.

¹⁶ For more complete information on industrial air conditioning, see *Heating, Piping, and Air Conditioning*: January, 1930, p. 8; December, 1930, p. 1026; March, 1931, p. 196; July, 1931, p. 553; also *Heating and Ventilating*, February, 1930, p. 90; March, 1930, p. 95.

In addition to (2), (3), (4), (5), and (6), above, the principal factors that fix the maximum cooling load are the maximum outside dry-bulb temperature and absolute humidity and the amount of radiant energy incident on the various building surfaces (principally all sunlit glass).

(b) The **building plans** and the local **Weather Bureau** are principal **sources** of the information necessary to estimate the maximum loads. The arrangement, extent, and nature of the materials used in the construction of the structure are available from the building plans. The direction and velocity of the prevailing winds, the maximum and minimum outdoor dry-bulb temperatures, and in some instances, the accompanying wet-bulb temperatures, which fix the absolute humidity, may be obtained from the Weather Bureau. Also, in very rare cases, the Weather Bureau may be equipped to supply readings of the intensity of the radiant energy received from the sun either on planes perpendicular to the sun's rays or on horizontal planes.¹⁷

(c) The **methods** used in estimating the maximum heating and cooling loads are partly empirical due to the uncertainty concerning some of the variables. The method of estimating the maximum heating load is standardized to a greater degree, at the present time, than that of determining the maximum cooling load. Only some of the more rational calculations will be given here, as the empirical estimates are not alike.

(d) The quantity of **heat transmitted through the walls** of a building depends upon the outdoor and indoor dry-bulb temperatures, the materials used in construction and their arrangement, and the wind velocity. The laws of heat transmission, given in Chapter XXVI, may be used in this calculation. The overall coefficient of heat transfer, U , for the building surfaces may be calculated from Eq. 427 (p. 283), when the materials of construction and their arrangement are known from the building plan or finished structure and when an average wind velocity is assumed. The film coefficient of heat transfer at the inside surface, h_1 , depends upon the nature of that surface and the air motion, but the values are commonly between 0.9 and 2.0 B.t.u. per hr. per deg. fahr. per sq. ft. of surface, and an average value for most calculations is 1.4. The film coefficient of heat transfer at the outside building surface, h_2 , is affected by the nature of that surface and the wind velocity. Based upon tests by Houghton and McDermott,¹⁸ the following empirical equations for h_2 are suggested:

¹⁷ See Heating, Piping, and Air Conditioning, Feb., 1932, p. 128, for description of a pyrliometer, the instrument used in measuring the intensity of this radiant energy.

¹⁸ "Heating, Piping, and Air Conditioning," March, 1931, p. 229.

For glass and painted wood, $h_2 = 1.4 + 0.28v$ (688a)

For concrete and brick, $h_2 = 2.0 + 0.4v$ (688b)

For rough stucco, $h_2 = 2.1 + 0.5v$ (688c)

where h_2 = the film coefficient of heat transfer at the outside surface,
in B.t.u. per hr. per deg. fahr. per sq. ft.

v = the wind velocity, in miles per hr.

The thermal conductivities of some building materials are given in Table XCVII. Additional information concerning other materials and built-up sections will be found in the current number of the A.S.H. and V.E. Guide, issued yearly.

TABLE XCVII

THERMAL CONDUCTIVITY OF BUILDING MATERIALS

Material	Specific Thermal Conductivity k	Thermal Conductivity for the Specified Thickness k/L
	B.t.u. per Hr. per Deg. Fahr. per Sq. Ft. for a Thickness of 1"	B.t.u. per Hr. per Deg. Fahr. per Sq. Ft.
White pine	0.78	
Yellow pine and fir	1.00	
Maple	1.10	
Glass	2.03	
Wood lath and plaster		2.00 (average)
Plaster (gypsum)	2.32	
Plaster board		3.73 per $\frac{3}{8}$ "
Cement mortar	8.00	
Hair felt	0.26	
Cork board	0.27-0.34 *	
Sugar cane or wood pulp fiber board	0.34	
Flaked gypsum	0.35-0.60 *	
Cellular gypsum	0.44-1.00 *	
6" hollow clay tile		0.54
8" hollow clay tile		0.89
Brick	5.00	
Stucco on wire mesh		8.00 (average)
Concrete (1 : 2 : 4)	8.30	
Asbestos shingles		6.00 (average)
Asphalt or composition roofing		6.50 (average)
Slate shingles	10.37	

* Increases as the density increases.

(e) The amount of air entering an enclosure by **infiltration** is very difficult to estimate. The usual method of estimating infiltration is to base the calculation upon laboratory tests of the infiltration through walls and windows. Many tests have shown that the infiltration of air *through walls* that are covered with building paper and properly plastered is substantially negligible when compared with the infiltration past cracks around *windows* and doors.¹⁹ Where double-hung wooden sash windows are installed in building, air gains entrance at the joints between the sash and frame (sash perimeter leakage), and also at the

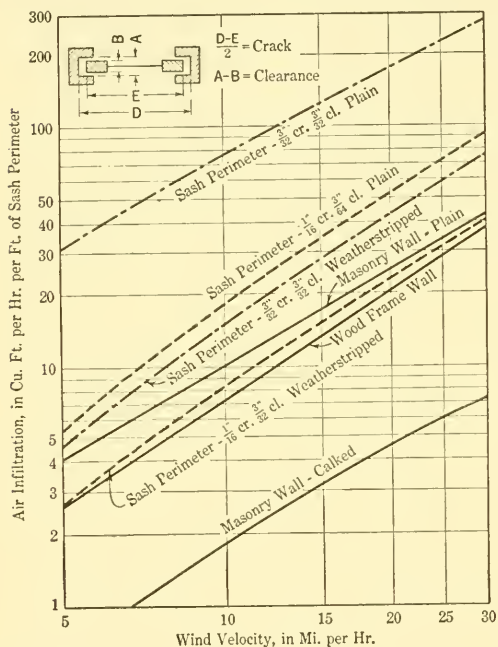


FIG. 1010.—Frame and Sash Perimeter Leakage for Double-Hung Wooden Sash Windows.

for masonry walls, plain and calked, as a function of wind velocity. The sash perimeter leakage is also shown for $\frac{1}{16}$ -in. crack and $\frac{3}{64}$ -in. clearance, plain and weatherstripped, and also for $\frac{3}{32}$ -in. crack and

frame openings (frame leakage). The sash perimeter leakage depends principally upon the wind velocity, the perimeter of the sash and the cracks and clearance of the sash. The meaning of crack and clearance is shown in Fig. 1010; these values depend upon the quality of the workmanship and materials used in the construction and the state of repair, but measurements of 579 plain double-hung wooden sash windows showed an average crack of $\frac{1}{16}$ in. and clearance of $\frac{3}{64}$ in.²⁰ The frame leakage, in cu. ft. of air per hr. per ft. of sash perimeter is shown in Fig. 1010 for wood frame walls and

¹⁹ See "Air Infiltration through Various Types of Brick Wall Construction," Heating, Piping, and Air Conditioning, Vol. 1, 1929, p. 585; also, "Air Infiltration through Various Types of Wood Frame Construction," Heating, Piping, and Air Conditioning, Vol. 2, 1930, p. 509.

²⁰ "Air Infiltration through Double-Hung Wood Windows," Heating, Piping, and Air Conditioning, Vol. 3, 1931, p. 583.

clearance, plain and weatherstripped. The data are taken from tests conducted at the University of Wisconsin in cooperation with the A.S.H. and V.E. Research Laboratory. The total leakage is the sum of the frame leakage and the sash perimeter leakage; for example, the total leakage through a double-hung wooden sash window set in a wood frame wall for a wind velocity of 15 mi. per hr. would be $(14 + 35)$ or 49 cu. ft. per hr. per ft. of sash perimeter for a $\frac{1}{16}$ -in. crack and $\frac{3}{64}$ -in. clearance without weatherstripping, or with weatherstripping $(14 + 16) = 30$ cu. ft. per hr. per ft. of sash perimeter.

The infiltration of air through metal sash windows is shown in Fig. 1011. The double-hung metal windows were tested as installed in the Southwestern Bell Telephone Building at St. Louis, while the residential casement and industrial pivoted windows, with other types of rolled section steel windows, were tested in the laboratory at the University of Michigan with the crack set as indicated on the curves.²¹ With metal windows the frame leakage is usually negligible, so the values in Fig. 1011 give the total leakage.

Assuming that the air pressure does not build up in an enclosure, the air that enters through some cracks must be leaving through others, so the total length of crack used in computing infiltration is the greatest crack in any one wall for a room with two or more exposed walls, except that in no case should less than one-half of the total crack be used. The length of the crack or leakage opening to be used in the computation is the total perimeter of the movable parts of the window. With a double-hung window this is always twice the height of the window plus three times the width. With the rolled section type of steel window with pivoted or swinging ventilator sections, the crack length varies with the area provided for ventilation. There are no data available on the infiltration through doors, but a fair estimate might be made

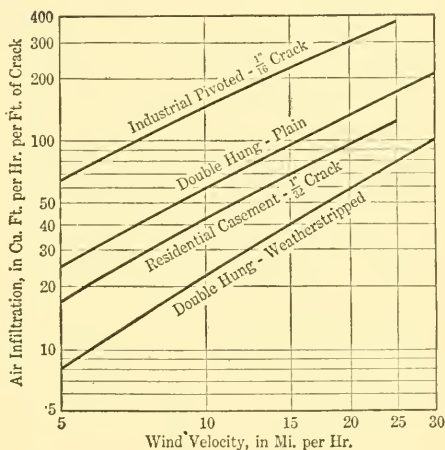


FIG. 1011.—Sash Perimeter Leakage through Metal Sash Windows.

²¹ "Air Leakage Studies on Metal Windows in a Modern Office Building." Trans. A.S.H.V.E., Vol. 34, 1928, p. 321; "The Weathertightness of Rolled Section Steel Windows," Trans. A.S.H.V.E., Vol. 34, 1928, p. 527.

by using the sash perimeter leakage in Fig. 1010 for the plain double-hung wooden sash window with a crack of $\frac{3}{32}$ in.

(f) The amount of **radiant energy** transmitted through window glass is a large part of the maximum summer cooling load. Due to the large heat storage capacity of the usual building walls, the effect of the sun upon the outside surface temperature and therefore upon the quantity of heat transmitted through the walls is small when compared with the effect upon window glass, since the glass transmits about 90 per cent of the incident energy. Rational methods, utilizing the principles of spherical trigonometry, may be developed to find the amount of radiant energy incident on horizontal and vertical planes in any locality at any time of the day and year, if the amount of radiant energy received in a plane perpendicular to the sun's rays is known.²²

With the above rational principles and a few empirical assumptions, the maximum heating and cooling load may be approximated.²³

²² See Heating and Ventilating: November, 1932, p. 14; June, 1933, p. 20.

²³ For a more complete discussion, see "Air Conditioning for Comfort," by S. R. Lewis.

CHAPTER LI

MECHANICAL REFRIGERATION

615. Definition.—By **refrigeration** is generally meant the removal of heat from a body or substance to such an extent as to leave it, or maintain it, at a lower temperature than that of its surroundings. **Mechanical refrigeration**, in the ordinary engineering application of this term, means the removal of heat by mechanisms or systems, the most important of which will be described in this chapter; however, the cooling of bodies or substances may also be done commercially in moderate climates by the use of ice, in very hot and dry climates by the naturally rapid evaporation of water, and in the laboratory by the use of liquefied vapors. These latter methods will not be discussed.

616. Basic Principles; the Carnot Cycle.—(a) The **function of a refrigeration system** is to maintain a cold body at a constant temperature lower than that of its surroundings by abstracting heat from the body at the same rate as the body absorbs heat. This abstraction of heat from the relatively cold body may be accomplished in a mechanical refrigeration system by the circulation of a suitable working substance, which is cold enough to absorb heat from the cold body during one process and hot enough to give up heat to a relatively hot body during another process. This second process cannot be made to follow the first one *directly*, but instead, there must be some intermediate process that requires the *expenditure of energy* upon the working substance to increase its temperature to a value at least equal to that of the hot body. The energy to operate a refrigerating machine is generally supplied directly as *mechanical energy*, but in some cases (such as in certain “absorption machines”) energy in the form of heat may be applied *directly* to operate the equipment. Many refrigerating plants are operated by heat-engines, such as steam engines and oil engines, and in these cases *heat* is *indirectly* used to produce *refrigeration*. When a refrigeration plant uses a compressor to raise the temperature of the working substance the desired amount, the plant is spoken of as operating on the *compression system*, as contrasted with the *absorption system*. As the compression system is by far the more important today, it will be considered first, and the other system will be discussed in Sect. 629.

Since the same working substance is used over and over again in the compression refrigeration cycle, not only is the compressor essential but also some suitable means must be provided to expand the fluid from the high temperature and pressure to the low ones. This expansion process will be explained in each of the various cycles involved.

(b) The **amount of mechanical energy required** to operate a refrigerating machine may be determined from the law of conservation of energy applied to the working substance, which completes one cycle after another. Thus, if the working substance has completed a refrigeration cycle in which there are *no losses due to leakage or friction, and no transfers of heat to or from the working substance except to and from the hot and cold bodies*, respectively, the following equations must hold for this *ideal cycle*:

$$Q_1 = Q_2 + E_m, \quad . \quad . \quad . \quad . \quad . \quad (689a)$$

or

$$E_m = Q_1 - Q_2, \quad . \quad . \quad . \quad . \quad . \quad (689b)$$

where

Q_1 = the quantity of heat delivered by the working substance
to the hot body,

Q_2 = the quantity of heat absorbed by the working substance
from the cold body,

E_m = the *mechanical energy supplied* to operate the machine.

These equations are exactly the same in *form* as those previously given ¹ for ideal heat engines, but note that the direction of the flow of energy in each case is just opposite to that for the engine.

(c) The **temperatures involved** in any actual or ideal refrigerating plant are always very important. The *temperature* (t_1) of the *hot body* is fixed by the natural conditions prevailing at the time and place where the refrigeration system is working; and this temperature is important because the hot body must always be able to absorb large amounts of heat from the working substance. This body is therefore generally some adequate supply of water such as a river, lake, or sea; but sometimes the atmosphere constitutes a more convenient form of the hot body, as with small household refrigerators. The *cold body temperature* (t_2), on the other hand, is fixed by the *purpose of the refrigeration*, such as the preservation of foods.

The *highest and lowest temperatures* of the *working substance* depend chiefly upon the values of t_1 and t_2 , respectively, but other important

¹ See Part I, p. 152.

factors are the *desired rates of heat transfer* that is to be maintained between the working substance and the hot and cold bodies, respectively. To keep the heat-transfer surfaces from becoming too large, and therefore too expensive, in any actual plant, the lowest temperature of the working substance must be somewhat below that of the cold body, and similarly, the highest temperature of the working substance must be somewhat above that of the hot body. The nearer the maximum and minimum temperatures of the working substance are to each other in any given case, the smaller is the mechanical energy required to operate the machine for a given amount of heat abstracted from the cold body; this is analogous to the water pump, handling a given amount of water, for the nearer the delivery and suction pressures are to each other the smaller is the energy required to operate the pump.

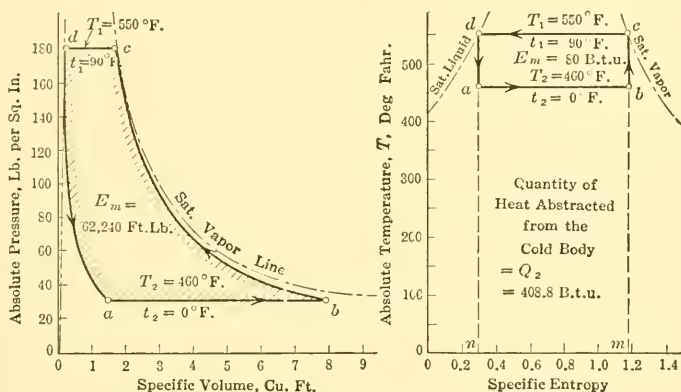


FIG. 1012.—The Carnot Refrigeration Cycle.

If the *rates of heat transfer* were not important, the maximum and minimum temperatures of the working substance could be made to approach, as limits, the temperatures of the hot and cold bodies, respectively. The nearer these limits are approached the nearer would the rate of heat transfer approach zero, and the nearer would such a cycle approximate the reversed Carnot cycle of a heat engine. A brief consideration of this cycle is therefore desirable because it shows the *limiting values* of the performance of a refrigerating plant working between the specified temperatures of the hot and cold bodies.

(d) The **Carnot cycle** for refrigeration is shown by *abcd* in Fig. 1012, the arrows indicating the direction in which the processes occur. Here a saturated liquid initially in state *d* is expanded *isentropically*² to the

² Note that such an expansion requires some form of an ideal engine, that may be either reciprocating or turbine.

state a , which is at the cold body temperature t_2 . Then the fluid is expanded *isothermally* from a to b (i.e., is vaporized at constant pressure), thus receiving from the cold body an amount of heat, Q_2 , represented by the area $abmn$. The fluid is now compressed *isentropically* from b to c , thus bringing it to the hot-body temperature, t_1 . Next the fluid is compressed *isothermally* (i.e., is condensed at constant pressure) as shown by the process cd during which the hot body receives from the fluid an amount of heat Q_1 , represented by the area $cdnm$. The fluid has now completed a Carnot cycle, but the *time* required would be infinitely long owing to the lack of any finite temperature head to cause a rapid transfer of heat during the two isothermal processes.

The mechanical energy, E_m , required from an outside source to carry out this cycle would be represented by the area $abcd$ on either diagram. Thus, on the $T\phi$ -diagram,

$$\begin{aligned} E_m &= Q_1 - Q_2 = \text{area } cdmn - \text{area } abmn \\ &= \text{area } abcd. \end{aligned}$$

The area $abcd$ on the pressure-volume diagram also represents E_m because this area is the excess of the work done *upon* the fluid above that done *by* the fluid during the cycle. In other words, E_m , or $abcd$, is the *net work* done on the fluid during the cycle. In this figure $E_m = 80$, and $Q_2 = 408.8$ B.t.u. per lb. of fluid when ammonia is the refrigerant used.

(e) The **coefficient of performance** (c.p.) of a refrigerating cycle is defined as the ratio of the heat abstracted from the cold body to the energy expended to attain this result.³ This coefficient is greater than unity, and the higher its value the better the performance. For any refrigeration cycle using the compression system, this coefficient is

$$\text{c.p.} = Q_2/E_m, \quad . \quad . \quad . \quad . \quad . \quad . \quad (690)$$

and for the Carnot,

$$\text{c.p.} = \frac{Q_2}{E_m} = \frac{\text{area } abmn}{\text{area } abcd} = \frac{T_2}{T_1 - T_2}. \quad . \quad . \quad . \quad (691)$$

Inspection of Eq. (691) will show that the value of the c.p. will become very large as T_1 approaches T_2 . In other words, the thermodynamic analysis indicates that the best performance of a refrigerating machine is obtained when the working substance is kept within the

³ Such a ratio might very properly be defined as the *thermal efficiency* of the cycle, but generally the term coefficient of performance has been used instead.

smallest possible temperature limits. It follows, therefore, that the body to be kept cool should be maintained at a temperature no lower than is necessary, and that the "heat receiver," or the hot body, should be at a temperature as low as nature furnishes at the particular time and place where it is needed. The marked effect upon the e.p. of the Carnot refrigerating system, produced by varying the two temperature limits, is shown by the curves in Fig. 1013. Clearly, comparative performances of different machines should be made only when they have the same values of t_1 , and the same values of t_2 .

The performance of a refrigerating machine may also be expressed by giving the amount of heat extracted from the cold body per unit of mechanical energy supplied. When this unit is the hp-hr. (2545 B.t.u.) it follows from Eq. (690) that the value of Q_2 , in B.t.u. per hp-hr. input, will be 2545 times the e.p.; and such values for the Carnot refrigerator may be obtained from Fig. 1013. When referring to any of the results in this figure, one should remember that no actual refrigerating plant will be able to do as well as the corresponding Carnot,⁴ which is independent of the kind of working substance used.

617. Gas and Vapor Refrigeration Cycles; General.—In the real refrigerating plant, decided advantages are obtained from the use of fluids that may be easily vaporized and liquefied at the temperatures

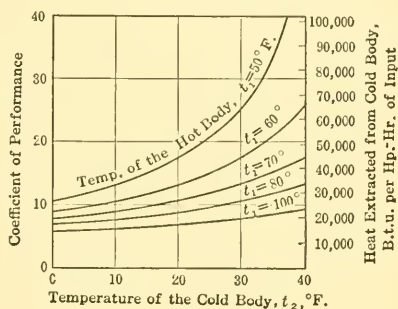


FIG. 1013.

involved in refrigeration work, instead of using those that remain a *gas* during the entire cycle. Many different kinds of vapors are used in refrigeration, and their relative merits will be discussed later. Air, because of its cheapness and other desirable characteristics, is the only fluid that has been used to any considerable extent as the working substance in a gas refrigeration cycle. The air cycle (first used in 1873) has now, however, been almost entirely superseded by the "vapor cycles" because the latter give better performances and use cheaper equipment which occupies less space; nevertheless a discussion of the air cycle will be given because a clearer understanding of the vapor cycles will be thus obtained.

The use of the term "vapor cycle" means that the working substance is a *vapor* while passing through that very important part of the

⁴ The corresponding Carnot strictly means a cycle having the same temperatures of the hot and cold bodies as those used in the actual plant.

equipment which involves the expenditure of mechanical energy, i.e., the compressor; however, in other parts of the cycle this working substance is a liquid or a mixture of vapor and liquid. Thus the short term "vapor cycle" serves in place of the much longer but more exact one, "the cycle of the working substance."

618. Air Refrigeration Cycles.—(a) Since air is non-poisonous, it can be brought into direct contact with foodstuffs and such, which are to be kept cool, without detriment to them. When such direct-contact cooling is used, the plant is said to operate on an *open cycle*; however, if the same mass of air is used continuously and repeatedly by circulating it through pipe coils instead of discharging it directly into the cooling chamber, the plant is said to operate on a *closed or dense-air cycle*. The essential equipment required for either kind of operation may be arranged as shown in Fig.

1014. Note that the air engine helps to drive the compressor.

(b) In an *open cycle*, air at atmospheric pressure is taken from the cold-storage room into the compressor cylinder and compressed to a pressure which may equal four or five atmospheres. The air is then delivered to the air cooler where its temperature is

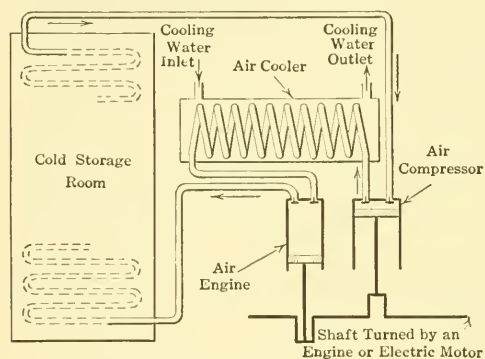


FIG. 1014.

materially reduced by allowing the air to flow around pipes carrying the coolest water available at a low cost. From the cooler the air passes, still under a high pressure, to the cylinder of an air engine. In this cylinder the air expands until its pressure again becomes atmospheric, thus delivering some mechanical energy to the compressor and at the same time being cooled to a temperature which may be of the order of -100 deg. fahr. This cold air is finally exhausted from the air engine to the cold-storage room where it absorbs heat. Thus, the contents of the cold-storage room are maintained at a low temperature by the removal of a small portion of the room air and the subsequent return of that air at a temperature much lower than that of the room.

(c) In the *closed or dense-air system*, the air is circulated through pipes or coils (shown dotted in Fig. 1014) in either a cold-storage room or a cooling chamber. In the latter case, some liquid, such as brine, is

usually circulated around the cold-air coils, and this liquid is then used to absorb heat from the cold body, located elsewhere. In this system, the delivery pressure of the expansion cylinder may be considerably above atmospheric, and by virtue of the higher pressures, the piston displacement, and thus the first cost, of both cylinders *per unit of refrigerating effect* may be decreased very materially; furthermore, the moisture may be removed once and for all from the air in this system, thereby preventing operating difficulties due to frost deposits. These advantages have given the closed-cycle machine the preference over the earlier open-cycle one, even though much more effective heat absorption is obtained with the latter system by bringing the cooling air directly into the cooling chamber, or into contact with the body to be cooled.

(d) The **ideal air cycle** is that shown by $abcd$ in Fig. 1015, in which the cycle of the working substance may be carried out by using machines of the reciprocating or other type.

For example, the compressor might be a centrifugal one and the air might expand in an air turbine; but regardless of the type used in the ideal cycle, the compression curve bc , and the expansion curve da , are both to be *isentropic*. Furthermore, the flow of the air into the cooling pipes (or into the cold room itself

when the direct system is used) is at the same rate as that from these pipes into the compressor; hence the pressure of the air in these pipes remains constant as shown by the line ab . Similarly the air passing through the air cooler is at constant pressure, as represented by the line cd . The amount of heat *abstracted from* the air in this cooler is represented on the $T\phi$ -diagram by the area $cdnm$; and the amount *absorbed by* the air in passing through the cold-storage room is shown by the area $abmn$. The area $lbckl$ on the PV -diagram represents the work done by the compressor *upon* the air during suction, compression, and delivery; and the area $kdalk$ represents the work done *by* the air in the engine during admission, expansion, and delivery. Thus, the difference between these areas on the PV -diagram, or the area $abcd$, must be the *net work* in foot-pounds required from an outside source to complete one ideal cycle of the air. It follows from Eq. (689*b*) that the area $abcd$ on the $T\phi$ -diagram also represents the same net work in B.t.u.

This information may be concisely expressed by means of equations.

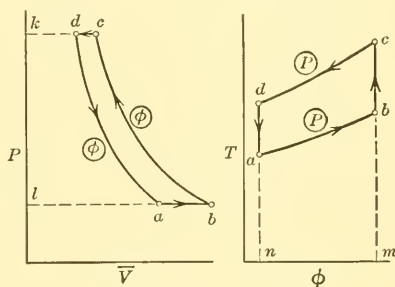


FIG. 1015.

Thus, if c_p is the constant-pressure specific heat of the air, the *quantity of heat removed from the cold room* by w pounds of air is

$$Q_2 = wc_p(T_b - T_a), \quad . \quad . \quad . \quad . \quad . \quad (692)$$

and that delivered to the air cooler (i.e., the hot body, which must be at a temperature low enough to reduce the air to an absolute temperature T_d), by w pounds of air completing the cycle is

$$Q_1 = wc_p(T_c - T_d), \quad . \quad . \quad . \quad . \quad . \quad (693)$$

and the *net work required* is

$$E_m = Q_1 - Q_2 = wc_p[T_c - T_d - (T_b - T_a)]. \quad . \quad . \quad (694)$$

Also, from the PV -diagram,

$$\begin{aligned} E_m &= \text{area } lbckl - \text{area } kdalk \\ &= w[(h_c - h_b) - (h_d - h_a)], \quad . \quad . \quad . \quad . \quad (695) \end{aligned}$$

in which h is the heat content, or enthalpy, of the air, in B.t.u. per lb. But

$$h_c - h_b = c_p(T_c - T_b), \quad \text{and} \quad h_d - h_a = c_p(T_d - T_a);$$

hence by these substitutions, Eq. (695) becomes the same as Eq. (694).

The *coefficient of performance* becomes ⁵

$$\text{c.p.} = \frac{Q_2}{E_m} = \frac{T_b - T_a}{T_c - T_d - (T_b - T_a)} = \frac{T_a}{T_d - T_a} = \frac{T_b}{T_c - T_b}. \quad (696)$$

If the c.p. is based on 1 hp-hr. of input, E_m in Eq. (696) becomes equal to 2545 B.t.u., and the *heat removed from the cold body, in B.t.u. per hp-hr.*, is

$$\frac{Q_2}{\text{hr-hr. input}} = 2545 \text{ c.p.} = \frac{2545 T_a}{T_d - T_a} = \frac{2545 T_b}{T_c - T_b}. \quad (697)$$

619. The Vapor Refrigeration Cycle.—(a) The **ideal vapor cycle**, like the air cycle, *might* consist of two constant-pressure processes and two isentropic ones; but it does not pay to try to utilize the small amount of mechanical energy available from passing a *liquid* through an engine. Consequently, instead of trying to use such an engine, an **expansion valve** is employed, and thus the liquid is **throttled** from a high pressure to a low one. The ideal vapor cycle, therefore, becomes that repre-

⁵ The last form of this equation is derived from the relation that

$$T_c/T_b = T_d/T_a, \quad \text{or} \quad T_c/T_d = T_b/T_a; \quad \text{hence} \quad T_c - T_d = \frac{T_d}{T_a}(T_b - T_a).$$

sented by $abcd$ in Fig. 1016. Here ab is the constant-pressure process that takes place in the *evaporator* while the mixture is absorbing heat. The vapor next passes through the *compressor* wherein its pressure and temperature are greatly increased during the process bc so that it becomes a superheated vapor in state c . The vapor is then delivered to the condenser, wherein condensation takes place at constant pressure as represented by the process cd . The resultant liquid is next passed through the *expansion valve*, and thus its pressure and temperature are greatly reduced so that it enters the evaporator in the state shown at a .

If the throttling from d to a takes place without heat transfer and so that the velocities at a and d are equal, h_a becomes known, for it is then equal to h_d ; but note that no line can be drawn to represent the *throttling process* between the known states d and a . However, a line, such as the broken one dga , may be drawn from d to a to represent the locus of

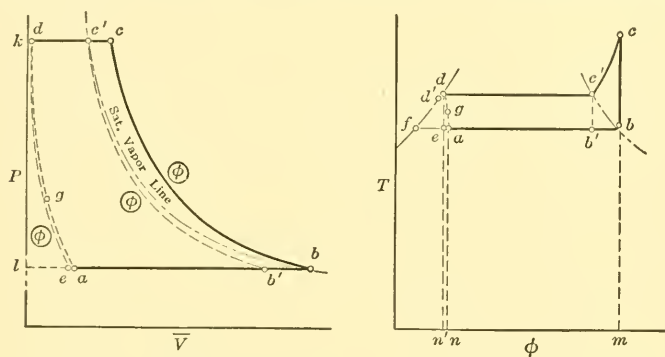


FIG. 1016.—Diagrams of an Ideal Vapor Cycle for Refrigeration.

successive states having the *same heat content* as at d ; but such a line is merely a convenient construction one and in no sense represents the continuous change of condition of the fluid during a throttling process. Consequently, the area enclosed by the heavy lines $abcd$ and the broken line dga *cannot* represent the net work of this ideal cycle. If an ideal engine had been used instead of the throttling valve the isentropic expansion line de would then result, and the small area $kdel$ ($= 778 (h_d - h_e)$) would represent the maximum amount of work obtainable from such an engine; and the net work of the cycle would then be represented by the area $abcdea$. Inspection of the figure clearly shows that the area $abcdga$ is less than $abcdea$; and this means that if $abcdga$ could really represent the net work, the throttling valve would have to be a device that delivers more useful work than an engine which permits isentropic expansion within it. Although such a conclusion is absurd,

there exists a false, but rather common, conception that the area $abcdga$ can represent the net work of the cycle.⁶

Work that might be obtained by passing the liquid through an engine instead of a throttling valve is clearly very small in the ideal case, and any real engine would not do as well; hence such a liquid engine is never used in the actual cycle, which is therefore based upon an ideal one having the irreversible throttling process.

(b) The **basic equipment needed** for the refrigeration vapor cycle is shown diagrammatically in Fig. 1017, and consists of four pieces, namely, the *evaporator*, the *compressor*, the *condenser*, and the *expansion valve*. For the *ideal* cycle the pipe lines connecting these four pieces of equipment are assumed to be *perfectly insulated*, and to have the fluid flow through them at such a slow rate that no appreciable drop in pressure is produced. In the *actual* plant, however, the insulation is imperfect, the flow has to be fairly rapid for economic reasons, and the

compressor has some friction and leakage; all these factors necessitate doing more work to drive the actual compressor than is required by the corresponding ideal. In the actual compressor, however, there is generally provided some means of cooling the vapor while it is passing through the compressor, and this tends to reduce the work required to

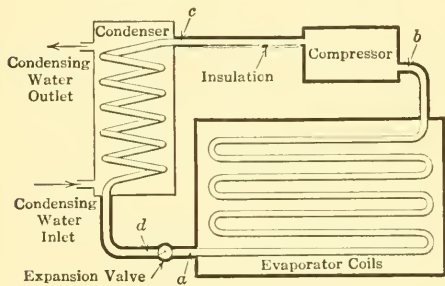


FIG. 1017.

a value less than that involved with the insulated machine; but this cooling effect is generally insufficient to overcome the other factors previously given, and thus the work required to operate the actual machine is greater than that of the corresponding ideal. This may also be expressed by saying that the coefficient of performance of the actual plant is less than that of the corresponding ideal.

(c) The **thermodynamic analysis of the ideal vapor cycle** may be readily made by reference to Fig. 1016. Using the notation previously given in this chapter, assume that *one pound* of the fluid is to complete the cycle; then, referring to the $T\phi$ -diagram, the mechanical energy required to operate the compressor is

$$E_m = Q_1 - Q_2 = \text{area } c d n' m - \text{area } a b m n . . . \quad (698)$$

$$= (h_c - h_d) - (h_b - h_a) \text{ B.t.u.} \quad (699)$$

⁶ For further discussion of graphical representations, see Sect. 55, Part I, p. 59.

But, as previously explained, $h_d = h_a$ in the ideal case, and thus the *work required*, in B.t.u., becomes

$$E_m = h_c - h_b. \quad . \quad . \quad . \quad . \quad . \quad . \quad (700)$$

This same result is apparent ⁷ also from the PV -diagram of Fig. 1016. Thus,

$$E_m = \text{area } bckl = 778 (h_c - h_b) \text{ ft.-lb.} \quad . \quad . \quad . \quad (701)$$

Eqs. (698) and (701) give *exact* values of the areas involved in the ideal vapor cycle regardless of what fluid may be used. *The area represented by Eq. (698) is very closely approximated by the area abcdfga* when the fluid is *ammonia*; but with carbon dioxide the error is somewhat larger because the pressure at *d* is much higher.

The *coefficient of performance* of this ideal vapor cycle is

$$\text{c.p.} = \frac{Q_2}{E_m} = \frac{h_b - h_a}{h_c - h_b}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (702)$$

The solution of Eqs. (701) and (702) becomes very simple when accomplished with the aid of a suitable vapor chart, such as that given for ammonia in Plate IV, in the Appendix. (The properties of ammonia are also given in Tables E and F, in the Appendix.

(d) When the **compression process** takes place wholly in the superheated region, as shown by *bc* in Fig. 1016, it is called **dry compression**; but when the vapor has a quality less than, or equal to, unity during its compression, the process is spoken of as **wet compression**. An illustration of wet compression in the ideal cycle is shown by *b'c'* in Fig. 1016. From an inspection of this figure one may observe that both Q_2 and E_m are reduced by using wet compression; furthermore, comparison with Fig. 1012 shows that the ideal cycle with wet compression approaches the Carnot more closely than that with dry compression; hence the coefficient of performance of the *ideal* cycle is slightly improved by operating with wet compression. However, the advantages of *dry* compression in the actual plant are now usually sufficient to make it generally the one preferred.

The **advantages of dry compression** are chiefly:

- (1) Higher volumetric efficiency of the compressor.⁸
- (2) Higher mechanical efficiency of the compressor, due to (1).
- (3) Less danger of damage to the compressor from the presence of liquid.

⁷ See Sect. 201, Part I, p. 315.

⁸ For a discussion of the factors affecting volumetric efficiency, see Power, Feb. 7, 1928, pp. 225-257.

(4) Higher compressor speeds with safety, and thus reduced first cost of the motor and compressor.

(5) Greater refrigerating effect per unit weight of fluid circulated through the system.

(6) Generally, slightly smaller power required to operate the compressor for the same refrigerating capacity.

The **advantages of wet compression** in the actual plant are:

(1) Less cooling water required for the same amount of refrigeration than when high superheat results from dry compression.

(2) Closer approach to the Carnot cycle.

(3) Less danger of dissociation of the vapor (such as ammonia), because the maximum temperature is lower.

(4) Easier to lubricate the cylinder when using ammonia, as a mixture of oil and liquid NH_3 clings to cylinder walls.

620. Capacity, Rating, and Performance of Refrigerating Machines.

—(a) When dealing with commercial refrigerating machines a large energy unit is needed, and the one selected by the American Society of Refrigerating Engineers (A.S.R.E.) is the **standard ton of refrigeration**, which is defined as **288,000 B.t.u.**⁹ It should be observed that no *time* element is involved in this term, which is a convenient *unit of energy* to use in refrigeration work; but unfortunately this *energy unit* is easily confused with another “ton unit” used to express the *rate* of refrigeration, because the A.S.R.E. has also defined the **standard commercial ton of refrigeration** as **288,000 B.t.u. per 24 hr.** (or 12,000 B.t.u. per hr., or 200 B.t.u. per min.). Consequently, in the commercial world, the capacity of the machine is generally specified simply by using the term *ton* without its qualifying term “commercial”; and in this sense a “10-ton machine” means one that is able to handle the working substance fast enough to produce 10 tons of refrigeration in 24 hr. The *capacity* of a refrigerating machine is naturally dependent upon the saturation temperatures corresponding to the pressures of the working substance entering and leaving it; hence certain temperatures have been agreed upon as satisfactory to use in specifying the standard rating of machines.

(b) The **standard rating** of a refrigerating machine¹⁰ using liquefi-

⁹ This number is approximately equal to the latent heat of fusion of 1 ton of ice. A more accurate value would be 2000×143.5 or 287,000 B.t.u.; but by taking the latent heat of fusion of ice as 144 B.t.u. per lb. the standard ton of refrigeration becomes the more convenient number given above.

¹⁰ In this definition a “refrigerating machine” means the compressor cylinder of the compression refrigerating system; or the absorber, liquor pump, and generator of the absorption refrigerating system.

able vapor is the number of standard commercial tons of refrigeration it performs under adopted refrigerant pressures. This definition is that given by the A.S.R.E., and the adopted pressures that have been specified by this society as standards for the purpose of rating a machine are:

(1) The **adopted inlet pressure** is that which corresponds to a saturation temperature of **5 deg. fahr.** (-15 deg. cent.) and is to be measured along the inlet pipe within 10 ft. of the machine.

(2) The **adopted outlet pressure** is that which corresponds to a saturation temperature of **86 deg. fahr.** (30 deg. cent.) and is to be measured along the outlet pipe within 10 ft. of the machine.

It is also further specified by the A.S.R.E. regarding these standards:

(1) that nothing but liquid shall enter the expansion valve; (2) that nothing but vapor shall enter the refrigerating machine; (3) that there shall be 9 deg. fahr. (5 deg. cent.) subcooling of the liquid entering the expansion valve, the point of measurement being within 10 ft. of this valve; and (4) that there shall be 9 deg. fahr. superheating of the vapor entering the refrigerating machine, the point of measurement being within 10 ft. of this machine.

(c) The **actual ice-making capacity** of a machine means the number of tons of ice the machine can make in 24 hr. This capacity is generally only from 50 to 75 per cent of its *ice-melting* capacity or "standard rating," because in actual ice-making, the water must first be cooled to 32 deg. fahr. before it is frozen, and after that the ice is further cooled to a temperature nearly the same as that of the brine, which may be at, say, 15 deg. fahr.—thus much less than a ton of ice usually results from the absorption of one standard ton of refrigeration. Furthermore, there are also heat transfers due to imperfect insulation that make the amount of heat absorbed by the working substance perhaps 10 per cent more than that abstracted from the water and ice.

(d) The **rate** at which the vapor must complete its cycle can be easily expressed in terms of the commercial ton, which means an absorption of heat at the rate of 12,000 B.t.u. per hr. Then, with the notation given in Sect. 619(c), it follows that the *weight* of vapor required to pass through the evaporator each hour to produce a commercial ton of refrigeration must be

$$w_v = \frac{12,000}{Q_2} = \frac{12,000}{h_b - h_a} \quad \cdot \cdot \cdot \cdot \cdot \quad (703)$$

The values of h_b and h_a clearly depend upon the fluid used, the pressure in the evaporator, and the state of the liquid before entering the expansion valve. Sometimes this liquid is a saturated one, as at d in Fig.

1016; but oftentimes the liquid is cooled to a temperature t_d below the saturation temperature corresponding to the condenser pressure, and it is then called a subcooled liquid; and in the latter case there would be a corresponding increase in the amount of refrigeration produced by a unit weight of fluid passing through the evaporator.

(e) The **performance** of any type of refrigerating machine, actual or ideal, may be specified by giving its coefficient of performance; but for machines of the *compression* type the general custom is to express the performance in terms of the **actual horsepower required per commercial ton of refrigeration**, which may be briefly expressed as (hp.)/(c. ton). When this power is merely that required to drive the compressor, the performance is that of the refrigerating machine alone; but if the performance is to include the mechanical losses in a reciprocating engine and in its driving mechanism, then the i.hp. of such an engine would be used in the previous expression; and if the compressor is driven by an electric motor the power input to this motor then becomes the item of most importance in expressing the performance.

The relation between the coefficient of performance and the *power* input is as follows:

$$\frac{2545 \text{ (hp. input)}}{\text{Rate of heat absorbed from cold body, in B.t.u. per hr.}} = \frac{1}{\text{c.p.}}$$

But since the standard commercial ton of refrigeration (c. ton) equals 12,000 B.t.u. per hr., it follows that the **power used per c. ton** is

$$\frac{\text{hp.}}{\text{c. ton}} = \frac{12,000}{2545 \text{ c.p.}} = \frac{4.71}{\text{c.p.}}, \quad (704)$$

or

$$\frac{\text{kw.}}{\text{c. ton}} = \frac{12,000}{3413 \text{ c.p.}} = \frac{3.52}{\text{c.p.}}, \quad (705)$$

The coefficient of performance of the Carnot refrigerating cycle for the standard temperatures ($t_1 = 86$ deg. and $t_2 = 5$ deg. fahr.) is readily found to be 5.74; hence by using Eq. (704) the corresponding value of the hp. per c. ton of refrigeration is found to be 0.821 for the Carnot having these temperature limits. The ideal vapor cycle with dry compression and the throttling process will not do as well as the Carnot because of these deviations therefrom, as may be observed from Fig. 1016.

The performance of an ideal or actual refrigerating machine may also be expressed by giving the amount of **energy, in hp-hr. or kw-hr., required to produce a standard ton of refrigeration (s.t.r.)** which is

288,000 B.t.u. The *energy* required to accomplish this amount of refrigeration with a machine having a known coefficient of performance, c.p., is

$$\frac{\text{hp-hr.}}{\text{s.t.r.}} = \frac{288,000}{2545 \text{ c.p.}} = \frac{113.3}{\text{c.p.}}, \quad . \quad . \quad . \quad . \quad . \quad (706)$$

or

$$\frac{\text{kw-hr.}}{\text{s.t.r.}} = \frac{288,000}{3413 \text{ c.p.}} = \frac{84.5}{\text{c.p.}}, \quad . \quad . \quad . \quad . \quad . \quad (707)$$

These results check those from Eqs. (704) and (705) by substituting for the s.t.r. its equivalent, (c. ton) $\times$ (24 hr.).

For the *actual machine* the coefficient of performance is appreciably less than that of the ideal using the same refrigerant and the same condenser and evaporator temperatures. With commercial machines the actual power delivered to the compressor is likely to be from 30 to 50 per cent greater than that of the corresponding ideal; and for small machines, such as household refrigerators, this percentage is much larger (see Sect. 632 (f)). The performances of the absorption systems will be discussed in Sect. 629 (g); and the ideal performances that may be expected from various refrigerants with the compression system are given in the following section.

621. Relative Advantages of Various Refrigerants.—(a) Although most of the large refrigerating machines use ammonia, this material is not the only refrigerant available. For plants used on shipboard, in restaurants and similar public places, and for domestic service, other refrigerants are often preferred. Many machines in stationary plants have been operated with carbon dioxide or with sulphur dioxide. Numerous other materials, including water, ethyl and methyl chloride, sulphuric ether, and dichloro-ethylene, have been used. The choice of a suitable working substance is based on many considerations; but those of volume and pressure are always very important.

(b) The **actual volume** of working substance required to cause a given amount of refrigeration determines the size of machine required. The larger the machine, the greater the friction losses and the first cost, other things being equal; hence bulk is undesirable because of the larger investment and the increased power required for operation.

The **pressure** is important in two ways. Some available substances have vapor pressures below atmospheric when at the temperatures common in refrigeration. Their use would mean the maintenance of a vacuum within the refrigerating machine, which is by no means a simple matter because of difficulty with air leakage. Other substances have vapor pressures so high that they can be used only with great difficulty.

A comparison of the pressures of the saturated vapors at 86 and 5 deg. fahr. for various refrigerants is afforded by the data given ¹¹ in Table XCVIII. This table also shows the volumes that must be handled by the compressor each minute for the same rate of refrigeration with these standard temperatures. Ammonia is seen to be quite satisfactory as regards both *bulk* and *pressure*. More than twice the bulk of sulphur dioxide is required for the same refrigerating effect. Carbon dioxide requires only about one-quarter the bulk of ammonia vapor, but it is not as convenient to use because of the very high pressures needed; however, it is cheaper than ammonia and is much less objectionable if it escapes. The pressure of water vapor is entirely below atmospheric at refrigerating temperatures, and that of sulphur dioxide is also below for the lowest temperatures and only slightly above for the highest temperatures.

For plants of large capacity either ammonia or carbon dioxide is commonly used. However, in all cases when choosing a suitable refrigerant, consideration should be given to its effect on the lubrication of the compressor, the danger from explosions, and its toxicity, as well as to its thermodynamic properties.

(c) The **coefficient of performance** of the ideal vapor cycle is greatly affected by the kind of refrigerant used, because the refrigerating effect and the work required per pound of vapor are both dependent on the thermodynamic properties of the fluid; the values of this coefficient for various refrigerants used in the ideal cycle with the standard saturation temperatures of 5 and 86 deg. fahr. are shown in Table XCVIII, which was calculated on the basis of no subcooling of the liquid before throttling and no superheating at the beginning of compression, except as indicated in the last column. The coefficient of performance of the Carnot cycle is independent of the kind of refrigerant; but for the ideal vapor cycle having dry compression and throttling through an expansion valve the properties of the refrigerant affect the performance, as may be seen from the values given in this table. In the actual plant, the performance is still further affected by fluid and mechanical friction, leakage, and imperfect insulation; and these factors are, to some extent, dependent upon the refrigerant used.

(d) The **amount of heat that can be absorbed** by a unit weight of fluid in passing through the evaporator coils is large for that substance

¹¹ This is an abridged and slightly modified form of the table given by W. H. Carrier and R. W. Waterfell in *Refrigerating Engineering*, June, 1924, p. 420. It should be recognized that some of the values given in this table are subject to considerable variation due to the incomplete knowledge of the thermodynamic properties of some of the refrigerants.

TABLE XCVIII

DATA REGARDING REFRIGERANTS FOR ONE COMMERCIAL TON OF REFRIGERATION, 5 TO 86 DEG. FAHR. (IDEAL CYCLES)

Cycle	Weight, Pounds per Minute	Volume, Cubic Feet per Minute	Initial Pressure at 5 Deg. Fahr.†	Final Pressure at 86 Deg. Fahr.‡	Ratio of Comp.	Coefficient of Per- formance	Horse- power	Relative Efficiency	Remarks
Carnot.....						5.74	0.8214	100.0	
Ammonia.....	0.4214	3.44	34.28	169.2	4.93	4.85	0.973	84.5	
Propane.....	1.396	3.35	43.70	159.0	3.64	4.88	0.9668	85.0	
Carbon dioxide.....	3.74	0.999	339.0	1054.0	3.11	2.56	1.843	44.6	
Carbon dioxide.....	2.091	0.559	339.0	1054.0	3.11	3.81	1.238	66.4	3 stage liquid cooling
Carbon dioxide.....	4.081*								
Carbon dioxide.....	2.625	0.702	339.0	1054.0	3.11	3.65	1.294	63.6	Liquid sub- cooled to 70 deg. fahr.
Sulphur dioxide.....	1.388	9.24	11.81	66.60	5.63	4.735	0.995	82.5	
Ethyl ether.....	1.555	60.8	3.52	25.05	7.12	4.86	0.971	84.6	
Dichloroethylene.....	1.768	108.4	1.78	14.65	8.23	5.14	0.918	89.4	
Carbon tetrachloride †..	2.532	307.0	0.53	5.62	10.58	5.09	0.927	88.6	
Trichloroethylene.....	2.137	513.0	0.315	3.42	10.84	5.085	0.928	88.5	
Water.....	0.1996	1972.0	0.0569	1.248	21.9	4.1	1.15	71.5	

* Weight in condenser.

† Carbon tetrachloride impracticable because too active chemically—dissociates and forms free chlorine.

‡ Pressures below atmospheric stated in inches of mercury.

‡ Pressures above atmospheric stated in pounds per square inch absolute.

having a *high latent heat of vaporization* at evaporator pressure and a *small heat content of the mixture* entering these coils. Since the heat content of the mixture entering the coils is equal to the heat content of the liquid before throttling, the smaller this value is in proportion to the latent heat of vaporization at the coil pressure, the better is this feature of the refrigerant for use in a vapor compression machine.

From this point of view, water is the best of the refrigerants previously considered, with ammonia next and carbon dioxide the worst, but the specific volume of saturated steam at 32 deg. fahr. is more than 700 times that of ammonia, and the pressure is extremely low (0.0887 lb. per sq. in. abs.). Furthermore, temperatures below 32 are generally needed; hence, water vapor is not commonly used in such cases, but it is used in some special applications, as explained in Sect. 628. When water vapor is the refrigerant used below 32 deg. fahr., the corresponding saturation pressures become extremely low, as may be seen from Table LXXXII, page 994.

(e) Many **new refrigerants** are being developed, and their properties can be determined accurately only by extensive and painstaking research. Some additional data regarding the properties of refrigerants are presented¹² in Table XCIX. For ammonia the properties are now well known, and are given in the Appendix by Plate IV and Tables E and F. For carbon dioxide and sulphur dioxide approximate values may be obtained from Plates VI and VII in the Appendix.

One of the relatively new refrigerants that is coming into considerable use is dichlorodifluoromethane, commonly called *freon* or *F-12*. The thermodynamic properties of this refrigerant are indicated by the Mollier chart¹³ given in Fig. 1018. This chart also shows by *a*, *b*, *c*, and *d*, the respective states of the fluid at entrance to the evaporator, the compressor, the condenser, and the expansion valve, for the ideal vapor cycle having the standard adopted inlet and outlet pressures for the compressor and 9 deg. superheat at *b* and 9 deg. sub-cooling at *d*. For this ideal cycle the coefficient of performance is 4.73; and if the fluid had been a saturated liquid at *d* and a saturated vapor at *b*, the coefficient would then be 4.71. These coefficients are about the same as those for sulphur dioxide, and slightly below those for ammonia. (See Table XCVIII.) Freon has a suction pressure of 26.5 lb. per sq. in. abs. for 5 deg. fahr., and is thus superior in this respect to SO₂, which has a saturation pressure less than atmospheric. Freon must be kept free of moisture when in contact with such metals as aluminum,

¹² Modified from the table given by H. D. Edwards, in *Refrigerating Engineering*, February, 1931, p. 112.

¹³ Plotted from the data given by A.S.R.E. Circular No. 12, 1931.

TABLE XCIX
SOME PROPERTIES OF REFRIGERANTS

Refrigerant	Symbol	Molecular Weight	Boiling Point at 1 Atmos.	Critical Temperature	Critical Pressure	Freezing Point	Liquid Density at Boiling Point	Gas Density *	Ratio of Specific Heats
			° F.	° F.	Lb./Sq. In. Abs.	° F.	Lb./Cu.Ft.	Lb./Cu.Ft.	$\gamma = c_p/c_v$
Dichloromethane.....	CH_2Cl_2	84.906	105.0	421.0	670 $\pm$	-142.0	186.9	0.225	1.14
Hexane.....	C_6H_{14}	86.112	155.9	454.6	435	-137.7	38.29	0.205	1.0
Pentane.....	C_5H_{12}	72.906	97.1	387.0	470	-203.4	38.09	0.185	1.0
Ethyl Chloride.....	$\text{C}_2\text{H}_5\text{Cl}$	64.497	54.5	369.0	764	-217.7	56.50	0.177	1.13
Butane †.....	C_4H_{10}	58.080	30.9	307.8	510	-211.0	37.51	0.1549	1.11
Sulphur dioxide.....	SO_2	64.064	14.0	315.0	1127	-97.6	91.15	0.1694	1.26
Isobutane.....	C_4H_{10}	58.080	+10.0	272.7	522	-229.0	37.20	0.1543	1.11
Methyl chloride.....	CH_3Cl	50.481	-11.4	289.6	954	-153.2	62.30	0.1336	1.20
Dichlorodifluoromethane §..	CCl_2F_2	120.916	-22 $\pm$			-247 $\pm$		0.338 $\pm$ †	1.12
Ammonia.....	NH_3	17.032	-28.0	270.3	1036	-107.9	42.56	0.0446	1.32
Propane.....	C_3H_8	44.064	-44.0	204.1	647	-309.8	36.35	0.1158	1.15
Propylene.....	C_3H_6	42.048	-53.7	196.1	645	-301.4	38.12	0.1087	
Carbon dioxide.....	CO_2	44.00	-109.3 †	88.0	1058	-69.9		0.1144	1.30
Ethane.....	C_2H_6	30.048	-127.5	90.1	694	-277.6	34.13	0.0782	1.22
Ethylene.....	C_2H_4	28.032	-154.7	+48.9	717	-272.9	35.49	0.0729	
Methane.....	CH_4	16.032	-253.9	-115.7	657	-297	19.20	0.0415	1.32

* At 30 in. Hg and 70° F.

† Solid.

‡ At 32° F.

§ Also called "F-12" and "freon."

brass, bearing metals, phosphor-bronze, and iron at temperatures as high as 212 deg. fahr.

Additional information regarding the properties of many of the new refrigerants are given in the A.S.R.E. Data Book.¹⁴

622. Compressors—(a) The compressor is the most important piece of equipment in the refrigerating plant, because the plant performance and cost of operation are most directly dependent upon it; also because it is the mechanism that is subject to wear at vital points which may permit leakage of the refrigerant and thereby possibly cause the injury or death of many people. Consequently compressors that are used for such fluids as ammonia have to be very carefully designed,

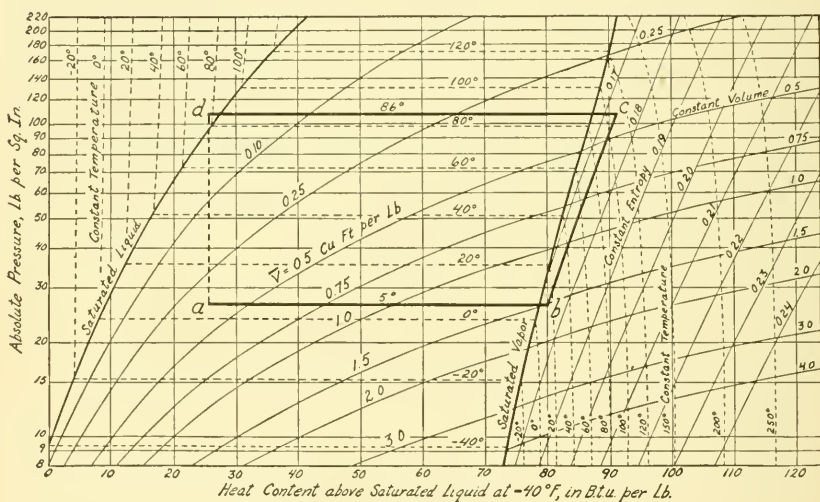


FIG. 1018.—Mollier Chart for Freon.

lubricated, and maintained. The types of compressors used in refrigeration are *reciprocating*, *gear*, *rotary* and *centrifugal*. For large plants the reciprocating type is by far the most common, but it is also used in

¹⁴ Also see Refrigerating Engineering as follows: "The Thermal Properties of Methyl Chloride," by D. N. Shorthose, August, 1924, p. 70; "Properties of Refrigerants," by H. D. Edwards, September, 1924, p. 95; "Thermal Properties of Sulphur Dioxide," by D. L. Fiske, December, 1924, p. 235; "Thermodynamic Properties of Butane, Isobutane and Propane," by L. I. Dana, A. C. Jenkins, J. N. Burdick, and R. C. Timm, June, 1926, p. 387; "New Methyl Bromide Data," November, 1931, p. 320; Mollier Diagram for CO₂, July, 1930, p. 32; Mollier Diagram for SO₂, January, 1931, p. 33; "Pressure—Total Heat Chart for Dichlorodifluoromethane," by Walter B. Lawrence, November, 1932, p. 286.

many small ones. However, with the new refrigerants and new applications of refrigeration, some of the other types are sometimes more suitable.

(b) **Reciprocating compressors** are generally used for ammonia and carbon dioxide because the delivery pressures are relatively high with these fluids. Such compressors are built as horizontal or vertical machines, and they may be single or double acting. When single acting they are often built with the connecting rod fastened to a piston pin; but many of the best designers believe that there should always be a cross-head to take the side thrust of the connecting rod and

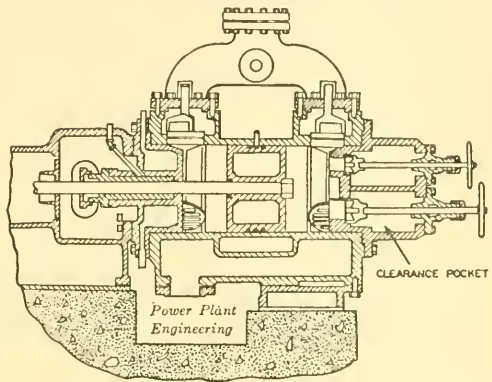


FIG. 1019.—Double-Acting Horizontal Compressor with Two Clearance Pockets on One End.

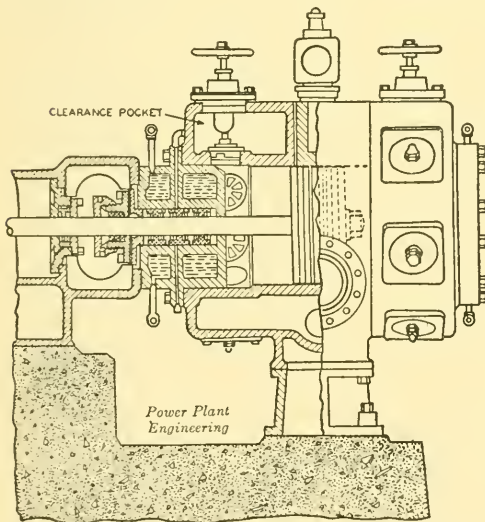


FIG. 1020.—Double-Acting Horizontal Compressor with Clearance Pockets on Both Ends.

thus prevent wear of the piston or cylinder. For ammonia compressors this is an important point because even slight leaks are sure to cause great annoyance. With the double-acting type of ammonia compressor the *stuffing box* has to be carefully designed and manufactured to prevent leakage around the piston rod. These boxes are generally made very long and are water cooled, as shown in Figs. 1019 and 1020. Very often, in the horizontal ammonia compressors, the stuffing box

contains two sets of packing separated by a small space called the *lantern*, which is kept full of oil under pressure.

(c) The **valves** of ammonia compressors are generally of the poppet

and plate types as in air compressors; but the tendency is in favor of the plate type because they can be operated at higher rotative speeds of the compressor with very little noise. The plate valves have the additional advantages of quick removal, examination, and replacement; and they have no protruding part that may break and fall into the cylinder. Sleeve valves¹⁵ are also coming into some favor in high-speed machines.

(d) The **speed** of large reciprocating ammonia compressors was formerly kept down to 100 r.p.m. or less; but higher speeds are now in common use and the tendency is to design the machines for still higher rotative speeds;¹⁶ because the first cost, weight, and space are thereby greatly reduced, and when they are directly driven by electric motors the lower cost of the motor is also of importance. By employing forced lubrication and plate valves, rotative speeds of 400 to 500 r.p.m. are now being used to some extent with machines having capacities of 25 to 50 tons per day.

The effect of speed on a 10½ in. by 18 in. simple, horizontal, double-acting ammonia compressor having a clearance of 2.7 per cent and plate valves (water jacketed) is shown by a test¹⁷ which gave the following results for a discharge pressure of 185 lb. per sq. in. gage and the three speeds shown:

Speed	Suction Pressure, Pounds per Square Inch Gage	Capacity, Commercial Tons	B.h.p.	B.h.p. per Commercial Ton
180	15	58.4	96.8	1.658
200	15	64.3	107.1	1.668
235	15	75.5	128.8	1.702
180	10	46.3	87.7	1.859
200	10	51.0	97.0	1.900
235	10	59.6	116.5	1.950

The entering temperature of the condensing water was about 68 deg. fahr. during these tests, which show that increasing the speed 30.5 per cent above 180 r.p.m. caused the capacity to increase 29.3 per

¹⁵ See "The Design of Compressors," by Bernard C. Oldham, Refrigerating Engineering, August, 1930, p. 83.

¹⁶ See "High Revolution Compressors for Marine Refrigeration," by E. Markham, Refrigerating Engineering, October, 1925, p. 111.

¹⁷ See "Effect of Speed on Compressor Capacity and Power," by J. Howard Jenks, Refrigerating Engineering, March, 1926, p. 291.

cent for a suction pressure of 15, and 28.7 per cent for a suction pressure of 10; and that the corresponding increase in the power required to drive the compressor per unit of capacity was only 2.7 per cent for the suction pressure of 15 and 4.9 per cent for the suction pressure of 10. The tests of this machine, when operated at a speed of 200 r.p.m. and a discharge pressure of 185 lb. per sq. in. gage, also give *volumetric efficiencies* as follows:

Suction pressure, lb. per sq. in. gage.....	25,	20,	15,	10,	5,	0.
Volumetric efficiency, per cent.....	78.8	76.4	73.4	69.7	64.2	55.4

For smaller machines the clearance is generally much greater than in this case (2.7 per cent); hence their volumetric efficiencies will be less than the above values. Sometimes the clearance is regulated to control the output of a machine driven at constant speed, as will now be explained.

(e) The **rate of delivery of a refrigerant** from a compressor should correspond to the desired rate of evaporation in the coils. When steam engines are used to drive the compressor, the speed may be easily controlled; but when constant-speed electric motors are used, other means of regulating the output of the compressor are required. This may be done by changing the number of compressor cylinders in use or by regulating the amount of clearance in the cylinder. With the latter method, there are several clearance pockets of different volume located in the cylinder heads, as shown in Figs. 1019 and 1020. By opening a pocket valve an additional clearance volume is introduced in the cylinder and a larger weight of vapor is thus trapped at the end of the discharge stroke; hence the reexpansion of this additional vapor to suction pressure will greatly reduce the volumetric efficiency. As a consequence, a smaller weight of fresh charge is taken into the cylinder, and the power required to operate the compressor is decreased, although both the speed of the prime mover and the coil pressure remain constant.

If a compressor with *fixed* clearance and a constant-speed driver were employed, when the quantity of heat absorbed by the vapor in the coils dropped below normal, the compressor would remove a greater weight of vapor than that produced by the coils per unit time; hence the pressure and temperature of the coils would drop until a new equilibrium condition was attained. As the coil pressure drops, however, the power required to operate the compressor per unit of refrigerating effect would increase. Thus, to give the best performance at all loads, the clearance pocket compressors are desirable when driven at constant-speed. However, the *best performance* of a machine is obtained with

a *small* constant clearance and some means of controlling its speed to suit the demands on the plant.

(f) The **water jacket** of a refrigerating compressor is not as effective as that on an air compressor, because the temperature of the refrigerant during the first part of the compression is generally lower than that of the water. Consequently, in most of the ammonia compressors, only the cylinder heads and usually about one-third of the cylinder are water jacketed. This water jacket has only a very slight effect upon the compression process but does prevent overheating of the cylinder and valves. In carbon dioxide compressors, the ratio of discharge to suction pressure is generally less than for ammonia compression, so for a given temperature of the vapor entering the compressor, the discharge temperature with CO_2 will be less than with NH_3 , and consequently the CO_2 compressors may be used without water jackets.

(g) **Multi-stage compressors** of the reciprocating type are employed when the compression ratio is very high. Just how high this ratio should be to justify the additional first cost of such a machine is somewhat debatable; but it is chiefly dependent upon the number of hours the plant is operated per year, and the annual savings effected in costs of operation. As a general rule, the compression ratio must be at least 5 or 6 before serious consideration need be given to the question.

The multi-staging of ammonia compressors is not as simple as in the case of air compression. In the first place, the discharge pressure of the ammonia compressor depends upon the temperature of the cooling water available for the condensers. Consequently, in the summer, the discharge pressure may have to be of the order of 160 lb. per sq. in. abs. or higher, whereas in the winter a discharge pressure of 100 lb. per sq. in. abs. may be allowable. Also, during the first part of the compression the temperature of the ammonia is considerably less than atmospheric. After the first stage of compression, no attempt is made to cool the vapor in the intercooler back to the temperature of the vapor entering the compressor, for this temperature may be of the order of 0 deg. fahr. or less. Instead, the vapor may be cooled to the temperature of vaporization corresponding to the intermediate pressure. If this pressure is 80 lb. per sq. in. abs., the temperature of vaporization is about 45 deg. fahr., and cooling to such a temperature could not be entirely effected economically with cold water. To secure these low temperatures, liquid ammonia is frequently used as the cooling medium, sometimes alone, and in other cases, after precooling the vapor with water.¹⁸

¹⁸ See Power, November 1, 1927.

When the compression of the vapor is effected in two stages, *two expansion valves* may be advantageously used between the condenser and coil. The first one permits expansion of the liquid to the intermediate compression pressure, as from state g to k for the ideal case shown in Fig. 1021. The dry vapor, which forms as a result of this expansion, may be separated from the liquid by means of a suitable separator and then passed to the suction line of the high-pressure compressor cylinder in some state such as that shown by d . The liquid remaining in the separator is now in the state d ; and after passing through the second expansion valve it enters the evaporator in the state a . The two advantages that have thus been gained by the two expansion valves are: (1) The work done by the low-pressure cylinder is less per unit of refrigeration produced because less weight of fluid is passed through this cylinder, and yet no appreciable reduction in refrigeration is caused by not passing the extra weight of *vapor* through the evaporator. (2) The evaporator coils are more effectively utilized, or in other words, less coil surface is required for the same refrigeration, because each unit weight of fluid passing through the coils absorbs the additional amount of heat represented by the area $aa'n'n$.

If cooling water is available at a sufficiently low temperature, an inter-cooler will enable the refrigerant to be cooled slightly below the temperature at c ; but to cool this superheated vapor down to the saturation temperature t_a

an intermediate tank or receiver is used, and into this tank *liquid ammonia* is injected with the superheated vapor. By this method of intercooling, the work of the high-pressure cylinder is greatly reduced, as shown by the area $cdec'$; but the liquid ammonia used for this intercooling has to be withdrawn from the condenser and thus the net gain is less than this area. An additional advantage obtained from the liquid injection is the avoidance of extremely high temperatures such as that at c' . Furthermore, when two refrigerator rooms are to be kept at different temperatures, the warmer one may often be supplied with liquid that has been throttled only to the intermediate pressure p_a , and thus the plant with two expansion valves and a two-stage compressor is particularly well suited for such applications.

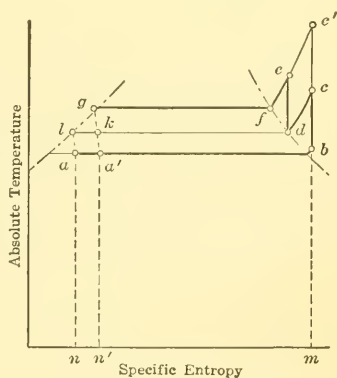


FIG. 1021.

A sectional view of a double-acting two-stage ammonia compressor of German design¹⁹ is shown in Fig. 1022. The low-pressure cylinder is placed between the two high-pressure ones, which are single acting; this enables a single stuffing box to be used and makes the machine very compact.

(h) **Rotary compressors** have been made in a number of different forms for many years, and have been employed for pumping fluids of several kinds. Their successful use in the refrigeration field is comparatively recent and has had to wait upon the development of a design which would be capable of operating at high speeds without the large friction losses and low volumetric efficiencies (especially in units of small capacity) which characterized previous apparatus of this general type.

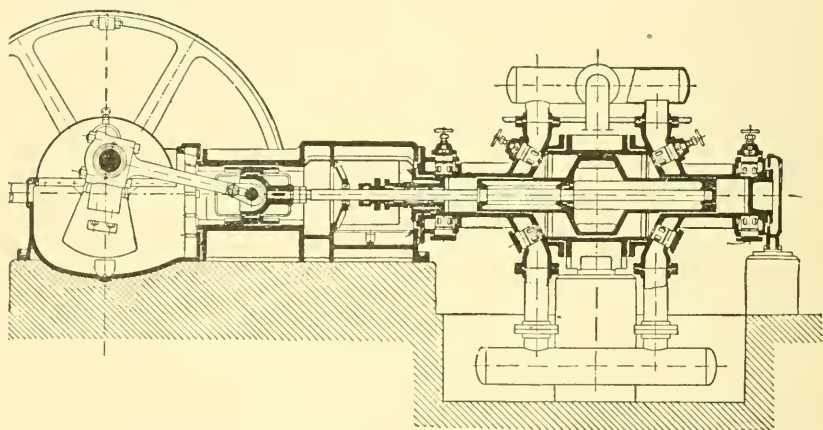


FIG. 1022.—A Double-Acting Two-Stage Ammonia Compressor Suitable for Large Capacities.

In some fields of service rotary compressors have been used for pumping against discharge pressures as high as 90 lb. per sq. in., with atmospheric pressure at the inlet, but such large pressure differentials between inlet and discharge are likely to cause excessive leakage of fluid past the contact surfaces between the moving elements and the casing; hence the successful employment of this type of machine in refrigeration work depends also upon the use of a refrigerant capable of operating with only a small pressure differential between the evaporator coils and the condenser.

All rotary compressors operate on the positive displacement principle

¹⁹ See "Compound Compressors for Refrigerating Plants," by Hans Freund, *Power*, Aug. 18, 1931, p. 232; also "Compound Compression," by the same author, in *Refrigerating Engineering*, April, 1933, p. 189.

whereby one or more elements, mounted on a rotating shaft, or shafts, and moving within a suitably designed casing, trap a definite volume of fluid during each revolution and carry it from the inlet to the discharge of the compressor, the arrangement of the devices being such that the leakage of fluid back to the inlet side of the casing is largely prevented. The compressor now used in the refrigeration field is of a unique design embodying features especially developed for the purpose of drastically reducing both friction losses and fluid leakage,²⁰ this device being well exemplified by the "rollator" of the Norge refrigerating machine which is explained in Sect. 632 (b) (see Fig. 1032). In this machine, leakage of refrigerant vapor from the high-pressure to the low-pressure side is reduced through the use of but a small pressure differential, a very accurate fitting of the parts with a compensation for wear, and the maintenance of an oil seal at the points of contact. During a routine manufacturing test, every "rollator" must show a volumetric efficiency of not less than 90 per cent when pumping air against a discharge pressure of 100 lb. per sq. in., and must give this result with a power consumption of not over 200 watts.

Compressors of the form just described are built in sizes for both household and industrial use; and they are so constructed that without changing the basic single-cylinder design, they may be arranged to operate as single-stage double-acting machines or as two-stage machines merely by the addition of suction and discharge valves, and of fixed blades, at the proper points around the periphery of the cylinder. Owing to their compactness they may be installed for multi-stage or booster service without requiring a large floor space, and advantage may thus be taken of the improved performance resulting from multi-stage operation in plants where very low temperatures must be maintained for special purposes.²¹

(i) **Centrifugal compressors** are sometimes used in vapor-compression systems of large capacity. These compressors, running at speeds of 3000 to 6000 r.p.m., are adapted particularly for direct drive by steam turbines, although electric motors are also sometimes used. The principal advantages of a centrifugal compressor of large capacity over a reciprocating one are its lower first cost, smaller space requirement, and the use of a refrigerant free from lubricating oil. The capacity required determines whether or not the centrifugal type is superior, since its efficiency is poor in small sizes. Centrifugal compressors have

²⁰ See "Rotary Compressors in Mechanical Refrigeration," by W. G. E. Rolaff, *Refrigerating Engineering*, November, 1928, p. 127.

²¹ See "Low Temperatures and the Rotary Compressor," by Harry Sloan, *Refrigerating Engineering*, June, 1932, p. 339.

been employed successfully in vapor-compression systems for air conditioning, the refrigerant used being dichloro-ethylene ($C_2H_2Cl_2$) or "dieline." This vapor is very dense and has low pressures at ordinary temperatures, condensing at 85 deg. fahr. with an absolute pressure of about 7 lb. per sq. in. The differential pressure between suction and delivery is of the order of only 7 lb. per sq. in. The centrifugal compressor may be successfully used with this refrigerant in machines with as small a capacity as 75 tons of refrigeration per day. If ammonia vapor is used as the refrigerant, it appears likely that turbine-driven centrifugal compressors are applicable only in capacities in excess of 300 tons per day.²²

(j) A **gear compressor** consists of a set of double-helical, or herringbone, gears which revolve within a closely fitted casing and which transfer the refrigerant vapor from the inlet to the discharge by means of positive displacement. The gears are mounted at their ends on double-ring ball bearings, and one of them is direct-driven by means of a motor, the usual speeds employed being either 1150 or 1750 r.p.m. Helical or herringbone teeth are used to effect a gradual discharge of vapor along the length of the teeth, thus preventing the occurrence of pulsations in the vapor discharge line. A constant supply of oil under pressure is furnished to the gears and bearings, and oil also fills the small spaces between the ends of the teeth and the casing during operation so that leakage of refrigerant from the discharge back to the inlet cannot occur at these points. Since the oil is carried out of the discharge with the vapor, a specially designed separator is used in the condenser to remove the oil from the vapor; and the lubricant is then cooled and returned to the compressor for use again. These compressor units have been built with capacities up to 60 tons of refrigeration per day; and they have the advantages of simplicity and compactness. High volumetric efficiencies have been obtained during tests of large-sized gear compressors, with power consumptions slightly over one horsepower per commercial ton of refrigeration.

(k) The **lubrication of the compressor** in a refrigerating machine presents a problem quite different from that found in air compressors. The temperatures encountered in refrigeration may vary from -70 deg. fahr. to 250 deg. fahr., which calls for a special grade of oil. This oil must be able to withstand low temperatures without congealing and high temperatures without vaporizing; also it must be free from animal and vegetable matter. Furthermore, the oil must be very anhydrous, since water carried over to the refrigerating coils will freeze, thus

²² "Turbo-Compressors for Refrigerating Purposes," by Dr. H. Voight, Power, Dec. 20, 1927.

decreasing the efficiency of heat transfer and eventually causing a shut-down. When SO_2 is the refrigerant, extreme care is necessary to use only a lubricant that is *entirely free* from moisture, because the combination of SO_2 and H_2O will produce a very corrosive acid. Ethyl chloride, methyl ether, and sulphuric ether are soluble in petroleum oils, and thus require the use of a lubricant such as glycerin, which does not absorb these refrigerants.

Oil vapor will be carried along with the refrigerant leaving the compressor, and unless removed, will pass into the condenser where it will collect on the walls of the tubes, and thus prevent effective transfer of heat. Hence an oil separator is connected to the discharge line of the compressor for the removal of as much of the oil as is possible. This separator should be located as far as possible from the compressor so that some of the oil vapor may cool and condense before it reaches the separator; and in some plants this condensation is hastened by passing the vapor through a cooler.

A *common type of oil separator* is a vertical cylinder with inlet and outlet openings near the top. In the cylinder are corrugated baffle plates suspended from the top. The vapor enters the separator at high velocity and strikes these baffle plates, the sudden change in direction tending to leave the oil clinging to the plates and the walls of the separator, from which it trickles down to the bottom and is drained off. The vapor, which is now nearly free from oil, passes around the ends of the baffles and flows to the condenser.

For small vertical piston compressors, the splash system of lubrication is used, while for large vertical compressors of the same type, as well as for all horizontal reciprocating compressors, the forced feed system of lubrication is generally employed. In order to keep the amount of oil in the refrigerant to a minimum, the quantity of oil fed to the compressor must be carefully regulated. In the splash system the level of the oil in the crank case is such that there is just enough splash for lubricating the parts; in the forced feed system the quantity depends upon the lubricating characteristics of the oil—the more favorable these characteristics the less oil required.

623. Refrigerant Condensers.—(a) The function of the condenser in a refrigeration system is to liquefy the vapor at as low a temperature and pressure as is practically possible with the available quantity and temperature of cooling water. If ammonia is the refrigerant, the vapor enters the condenser at temperatures ranging from 200 to 300 deg. fahr. and at pressures of 100 to 200 lb. per sq. in. abs. The circulating water must then absorb from each pound of vapor an amount of heat equal to the difference between the heat content of the entering superheated

vapor and the heat content of the resulting saturated liquid, both at condenser pressure. The development of the ammonia condenser has proceeded with more consideration given to safety than to efficiency of operation; and hence from the standpoint of performance, has been developed more slowly than the steam condenser, although the heat-transfer process involved is similar.

The **types of condensers** commonly used in refrigeration may be divided into four distinct classes: (1) submerged-coil; (2) atmospheric; (3) double-pipe; and (4) shell-and-tube.

(b) The **submerged-coil condenser** was the type first to be used in refrigerating systems. Such condensers consist essentially of a coil of pipe placed in a tank of water. The refrigerant enters the pipe at the top and flows downward through the coil, while the cooling water enters the tank at the bottom and overflows at the top of the tank. Although this condenser is safe, cheap, and easy to build, the heat-transfer coefficient is extremely small because of the low water velocities used; also, since only a small part of the water is in direct contact with the coil, but little heat is removed per gallon of water circulated.

(c) The **atmospheric condenser** was the next step in the development. This condenser consists of horizontal pipes arranged in one or more vertical rows, or stands, with the pipe ends connected by return bends so as to form a continuous path for the flow of the vapor. The cooling water is fed into a trough over each row and drops upon the surface of the top pipe, from which it splashes and drips upon the succeeding pipes, eventually being collected in a catch basin under the bottom pipe. Atmospheric condensers may be subdivided, according to the direction of vapor flow, into three types, viz.: *common*, *bleeder* or *counterflow*, and *flooded*. (1) In the **common atmospheric condenser** (Fig. 1023), the vapor enters at the top and leaves at the bottom, thus there is parallel flow of the refrigerant and cooling water. (2) In the **bleeder or counterflow atmospheric condenser** the vapor enters at the bottom and leaves at the top. The advantage of counterflow over parallel flow in condensers is not so marked as it is in some other forms of heat-exchange apparatus, because the greater portion of the heat transfer occurs while the vapor remains at the constant condensing temperature; as far as the heat-transfer rate is concerned, other factors being the same, it makes no difference in which relative directions the fluids are flowing when one fluid remains at a constant temperature. This type is called the "bleeder condenser" because some of the condensate is removed by draining several of the upper pipes, as in Fig. 1024, thus preventing the flow of most of the liquid refrigerant to the bottom pipes where partial re-evaporation may occur. (3) In the

flooded atmospheric condenser, a nozzle is placed in the vapor line about three pipes up from the bottom and is so connected that, as the vapor flows through it, an injector action is set up whereby some of the condensate from the bottom pipes is drawn into the stream and again passed through some of the pipes. Belief that the presence of liquid refrigerant in the vapor would increase the rate of heat transfer prompted

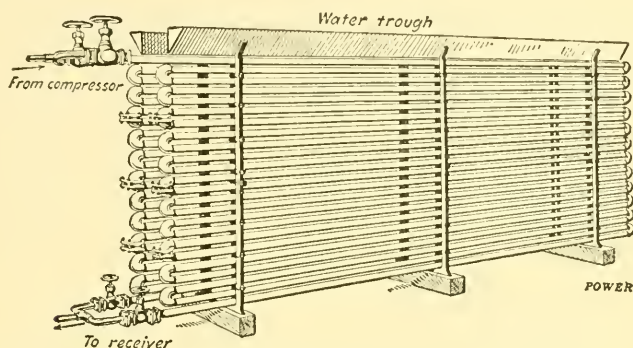


FIG. 1023.—Common Atmospheric Type of Ammonia Condenser.

this design. Reliable tests²³ show, however, that the best performance of a condenser is obtained when the condensate is removed completely and immediately from the condensing surface.

An atmospheric condenser cannot be forced to greatly increased capacity by using more than the normal amount of water. A condenser of this kind, however, is particularly suitable when the cooling water is bad and the space available is not restricted; and it is especially applicable when the condenser is to be located on a roof.

(d) The **double-pipe condenser** was developed to enclose the flowing water in order to prevent splashing, and to provide for close regulation of the water velocity and therefore of the rate of heat transfer. This condenser is made up of one or more vertical rows of concentric pipe. In the usual arrangement, the cooling water flows up through the inner pipe, while the vapor flows down through the annular space between the pipes, the respective passages being

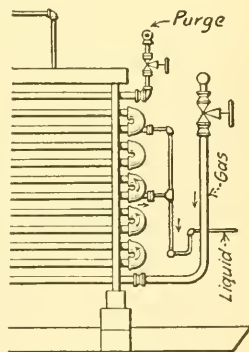


FIG. 1024.

²³ "Heat Transfer in Ammonia Condensers," by Kratz, Macintire, and Gould, Bul. 171, Univ. of Ill. Engg. Expt. Station.

connected in series (see Fig. 1025). This type of condenser is very flexible as to operation, and, if high water velocities are used, a large capacity can be installed in a comparatively small space.

(e) The **shell-and-tube condenser** is the latest type to be developed and is being rather widely adopted. This type of condenser is quite similar to the steam surface condenser with tube ends rolled into both tube sheets. The tubes may be either vertical or horizontal and confined in a single shell or in multiple shells, and may be arranged to provide a single pass or multi-pass for the water.

In the **vertical single-pass single-shell arrangement**, the water enters at the top and is uniformly distributed to the tubes by means of a baffle placed above the tube sheets. On the top end of each tube there usually are water-distributing devices which may have spiral grooves to give the water a whirling or corkscrew flow down the inner tube

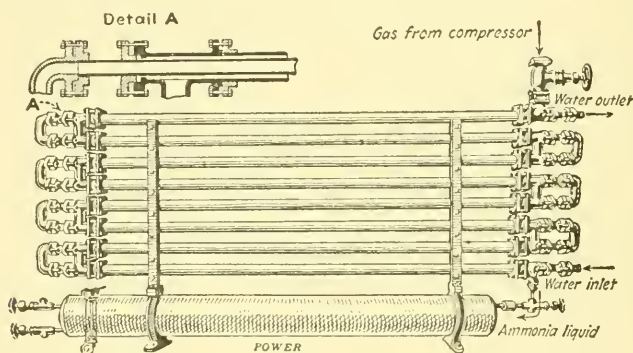


FIG. 1025.—A Double-Pipe Counterflow Condenser.

surfaces. The object of this is to maintain a thin, rapidly moving water film in close contact with the tube surface. The vapor enters either at the top or mid-section of the shell, and its condensate flows downward over the tube surfaces to the liquid outlet pipe at the bottom of the shell. A condenser of this type may also be built with single horizontal shell, and then it is generally of the multi-pass type similar to the multi-pass steam surface condenser.

A horizontal **multishell-multitube condenser** of one form is shown in Fig. 1026. This condenser was constructed for experimental purposes and has only two shells, the tubes in each of which are so arranged that the water passes through them in series, the pipe ends being connected by return bends. The water circuit shown provides for operating the elements in parallel; the arrangement can be readily modified, however, to have series operation. In still another arrangement the

tubes extend between tube plates in each horizontal shell, as in the single-shell construction, and large water return bends connect the shells in series.

Multi-tube condensers occupy the least space of any type for given capacity, have the fewest joints to maintain tight, and require considerably less power for water circulation than the double-pipe condenser.

(f) In all the condensers, the presence of scale or other deposits on the tube surfaces, and the presence of non-condensable gases in the vapor, will materially decrease the effectiveness of the surface; so provision must be made for frequent cleaning of the tubes and for removal of inert gas through flow off or "purge" valves and piping. If the cooling water must be conserved, a cooling tower or pond can be included in the condensing systems and it is often placed on the roof. The materials exposed to a refrigerant must be such as will not be

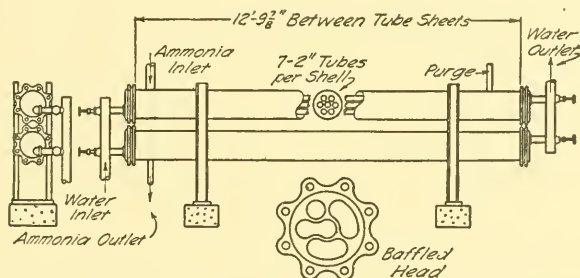


FIG. 1026.—Experimental Multishell-Multitube Condenser.

deteriorated thereby; for example, ammonia attacks brass and other copper alloys, but has no effect on iron or steel. The number of joints that may cause trouble should be reduced as much as possible, as by welding.

(g) The performance characteristics and overall coefficients of heat transfer of ammonia condensers of the (1) atmospheric-bleeder, (2) double-pipe, (3) vertical shell-and-tube, and (4) multitube-multipass horizontal shell-and-tube types have been quite thoroughly investigated by the Engineering Experiment Station of the University of Illinois.²⁴ The determination of the proportions of new condensers of any of these types for given conditions can be based on the data given in the bulletins mentioned in the footnote.

²⁴ Bulls. 171, 186, and 209, "Heat Transfer in Ammonia Condensers," by A. T. Kratz, H. J. Macintire, and R. E. Gould.

These investigations show that, for an initial water temperature of 68 deg. fahr. and a condenser pressure of 159.7 lb. per sq. in. abs. (ammonia saturation temperature of 82 deg. fahr.) the performance of all four types of condensers, expressed in terms of the amount of heat removed from the ammonia per hour per square foot of condenser surface, may be represented as a linear function of the gallons of condensing water used per minute per square foot of condenser surface by the empirical equation

$$q/A = 5100G + 225, \quad . \quad . \quad . \quad . \quad . \quad (708)$$

in which q/A = the quantity of heat removed from the ammonia, in B.t.u. per hour per external square foot of condenser surface, and G = the rate of water circulation, gallons per minute per square foot of external surface. This is for clean tubes.

The actual rates of heat transfer for any of the four condensers did not deviate by more than 8 per cent from the values given by this equation over the range of conditions covered by the tests. The departure of the water temperature from 68 deg. fahr., however, was found to have quite an influence on the value of q/A , other conditions being the same; for example, with a liquefaction pressure of 145 lb. per sq. in. and a water flow of 0.22 gal. per min. per sq. ft., these heat transfer rates at temperatures 72, 68, 64, and 60 deg. fahr. were found to be related to each other in the ratios 8 : 11 : 14 : 18, respectively, the values for 68 deg. being about as given by Eq. (708). The overall coefficient of heat transfer (U), as estimated by the methods given in Sect. 338, page 282, and as illustrated by the example given on page 284, check closely with the overall coefficients obtained in the tests when allowance is made for fouling.

These tests also indicate that the best design for a condenser is one in which the condensate is completely removed from the condensing surface immediately after condensation, and that the most effective arrangement of the condensing surface is one in which the length of vapor travel is a minimum and in which a maximum amount of surface is exposed to the coldest water. It also appears that a certain portion of surface in most of the condensers is relatively ineffective, and that therefore it may be possible to reduce materially the amount of surface over that now installed for a given capacity.²⁵

²⁵ Some additional references on refrigerant condensers: "Ruling Factors in the Selection of an Ammonia Condenser," by L. C. Miller, in *Power*, Nov. 7, 1922, p. 721; "Operating Characteristics of Ammonia Condensers," by W. H. Motz, in *Power*, June 29, 1926, p. 1012; "Modern Ammonia Condenser Design," by W. R. Kitzmiller, in *Power*, April 19, 1932, p. 592.

624. Precoolers and Accumulators.—(a) When the condensate leaves the condenser it is substantially at the saturation temperature corresponding to the condenser pressure, and this temperature is likely to be from 10 to 20 deg. fahr. higher than that of the condensing water entering the condenser. If this liquid is cooled by passing through a device called a **precooler**²⁶ before going to the expansion valve the refrigerating effect produced by each unit weight of fluid passing through the evaporator will be greater. A small amount of precooling may be produced by using the condensing water as a cooling medium before it goes to the condenser. The weight of condensing water is likely to be from 60 to 80 times as great as that of ammonia; and since the specific heat of liquid ammonia is only from 10 to 15 per cent greater than that of water, the rise in the temperature of the water in passing through a counterflow type of precooler will be relatively small. Sometimes a limited amount of unusually cool water is available for use in the precooler and then a much lower temperature of the liquid may be obtained, or in other words, the *subcooling* of the liquid is better. At best, however, the amount of the precooling that can be accomplished by water alone is generally very small, but a still greater amount is often obtained by other means, one of which is by using a combined precooler and accumulator.

(b) An **accumulator**, as used in a refrigerating plant, is basically a separator, to keep *vapor* from entering the evaporator and to keep any *liquid* from entering the suction line of the compressor. The connections to and from an accumulator are shown in Fig. 1027 (a); and, as long as liquid fills the lower part of the accumulator, nothing but liquid can enter the coils of the evaporator, which are therefore said to be *flooded*. Such a term is somewhat misleading because the quality of the fluid in the coils gradually becomes higher and higher as the fluid progresses through them. The accumulator also acts as a small *reservoir* in which the liquid accumulates at the bottom, and then flows by gravity to the coils.

The accumulator will also act as a *precooler* when supplied with cooling coils through which the liquid passes before going to the expansion valve, as shown in Fig. 1027 (b). This will give very effective precooling because these coils are surrounded by the vapor and liquid at the temperature of the evaporator. However, this extra precooling has been obtained by the evaporation of some additional vapor that must be handled by the compressor even though it did not come from the evaporator coils. With compound compressors and two expansion

²⁶ The term *aftercooler* is also used because the liquid is being cooled *after* leaving the condenser.

valves,²⁷ however, this method of precooling is especially advantageous.

625. Expansion Valves and Control Systems.—(a) All expansion valves are basically throttling devices, and the throttling action has been explained in Sect. 619 (a); but these valves also serve to *control the rate of flow* of the liquid to the evaporator so that the desired rate

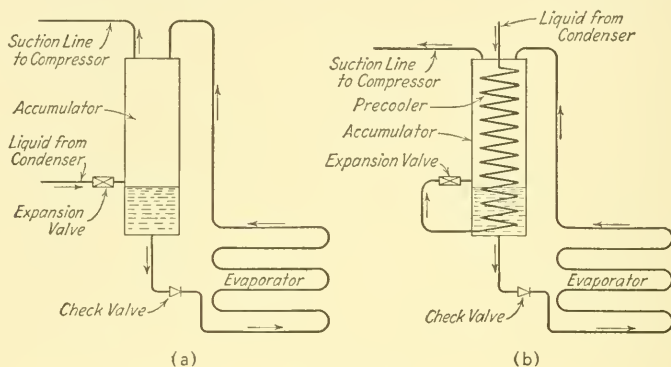


FIG. 1027.

of refrigeration may be maintained. Expansion valves are either hand-controlled or automatic. An illustration of a **manually controlled valve** is shown in Fig. 1028. When valves of this type are employed the operator has to be constantly alert to keep the plant operating so that the pressure in the evaporator will be no lower than necessary and so that only dry vapor enters the compressor. A few degrees of superheat in this vapor is far preferable to slugs of liquid that might wreck the compressor.

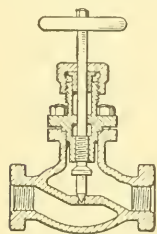


FIG. 1028.

(b) **Automatic valves** are now very extensively used in large refrigerating plants, and naturally they are always used in household refrigerators. Automatic valves are actuated by various means, such as the suction line pressure, condenser pressure, condenser temperature, evaporator temperature, or by some combination of them.

When a plant is under full automatic control, it should have a suitable device to start and stop the prime mover, another to regulate the opening of the expansion valve, and a third to control the rate of flow of the condensing water. Naturally the perfect control of all these factors requires more than can be expected of automatic devices. On the other hand, by having the operator

²⁷ See Sect. 622 (g).

make occasional adjustments, the automatic control will give better results than can be obtained solely with a manually operated valve. Some examples of automatically controlled valves are given in Sect. 632 for household refrigerators.

626. Evaporators.—(a) The function of an **evaporator** in a vapor refrigerating system is to provide for the transfer of heat from the cold chamber, or the body being cooled, to the refrigerant which is thereby converted from a liquid to a vapor. An evaporator consists essentially of a series of pipe coils through which the refrigerant flows, the general shape and arrangement of these coils depending upon the design of the cooling chamber, the nature of the bodies or substances being cooled, the desired regulation of the flow of refrigerant, and the allowable pressure drop through the coils.²⁸ Evaporators may be operated on either the *flash system* or the *flooded system*, the proper designation depending upon how this apparatus functions in connection with other pieces of equipment in the plant.²⁹

(b) With the **flash system** the refrigerant is discharged from the expansion valve into the coils of the evaporator, while the exit end of the evaporator is connected directly to the suction line leading to the compressor. A portion of the liquid refrigerant flashes into vapor upon leaving the expansion valve, since the pressure in the evaporator is less than the saturation pressure corresponding to the temperature of the liquid, and the energy absorbed during this action cools the remaining refrigerant to some extent. The result is that the coils nearest the expansion valve will contain considerable vapor which reduces the rate of heat transfer from the metal of the coil to the refrigerant; furthermore, this flashed vapor is not capable of removing heat from the cold chamber, yet it must be carried through the entire coil system on its way to the compressor. When the flash system of evaporation is used, the expansion valve is adjusted by hand or by thermostatic devices in order that the proper rate of flow of refrigerant may be maintained.

(c) The **flooded system** operates in conjunction with an accumulator or a combined accumulator and precooler, the arrangement being similar to those shown in Fig. 1027. In this case the first coil of the evaporator is always filled with liquid refrigerant which is completely available for the absorption of heat through its evaporation; and furthermore, the

²⁸ See "The Design of Refrigerator Evaporators," by D. P. Heath, Refrigerating Engineering, July, 1931, p. 27; also see "Modern Refrigeration Evaporating Systems," by W. H. Motz, Ice and Refrigeration, March, 1931, p. 269.

²⁹ See "Factors Affecting the Design of Evaporators," by H. Vetter, Refrigerating Engineering, December, 1932, p. 348.

rate of heat transfer from the metal pipe to the liquid is considerably higher than that from the pipe to vapor, as in the flash system. With the flooded system the rate of flow of liquid refrigerant to the evaporator may be accurately controlled by some kind of *float valve* installed in connection with the accumulator (see Sect. 625). The flooded system is preferred in most plant designs because of its more effective use of the refrigerant as well as its superior ability to maintain the desired temperature under close control.

Various modifications of the evaporator systems just described are often used,³⁰ but they cannot be treated here. In all modern designs increased attention is being given to the improved thermal performance of the evaporator system; and this includes studies of the **coefficients of heat transfer** for the various arrangements of equipment employed. The values of these coefficients cover a wide range, from a value of about 2 B.t.u. per hr. per sq. ft. per deg. fahr. when the flash system is used for cooling air, to a value of 15 when the flooded system is used for cooling a liquid.

627. Dry Ice.—(a) **Solid carbon dioxide**, or **dry ice**, is produced when liquid CO_2 expands from a high pressure to a pressure for which the saturation temperature is below its freezing point (-70 deg. fahr.). Formerly, solid CO_2 was produced principally in laboratory experiments designed to demonstrate the properties of solids, liquids, and vapors, but at the present time, the production of solid CO_2 is of considerable commercial importance.

(b) The **production of dry ice** involves (1) the formation of gaseous CO_2 of suitable purity, (2) the compression and condensation of the CO_2 , and (3) the expansion and consequent freezing of the liquid. The principal commercial sources of gaseous CO_2 are from the combustion of coke, the calcination of limestone, and as a by-product of the alcohol and chemical industries.

In the coke process, the coke is burned in a boiler furnace, the products of combustion supply the CO_2 , and the heat developed in combustion generates steam that may be used for power. The products of combustion are scrubbed and cooled, then brought into contact with a solution of potassium carbonate or sodium carbonate which absorbs some of the CO_2 ; the bicarbonate solution is then pumped to the generator where the CO_2 is driven off by heating with steam coils. The CO_2 is then cooled and the condensed water vapor removed before the CO_2 vapor is compressed in several stages with intercooling, to pressures between 1000 and 1500 lb. per sq. in. By this means the CO_2

³⁰ See "Modern Refrigeration Evaporating Systems," by George Hilger, *Refrigerating Engineering*, October, 1928, p. 99.

vapor is condensed and the liquid is then expanded through nozzles into evaporators. The solid CO_2 thus produced is removed and pressed into blocks while the vapor formed during the expansion is returned to the compressor through a heat exchanger.

(c) The **chief advantages** of solid carbon dioxide as a refrigerant, when compared with ordinary ice, are: (1) the greater refrigerating effect per pound, (2) the lower temperature of sublimation, and (3) the formation of a vapor instead of a liquid as the refrigerant absorbs heat. The sublimation of solid CO_2 , from the saturation temperature of -110 deg. fahr. at atmospheric pressure to superheated vapor at 0 deg. fahr. and the same pressure, requires an addition of heat of about 274 B.t.u. per lb. as compared with the latent heat of fusion of 144 B.t.u. for ice at 32 deg. fahr.

(d) The **chief applications** of dry ice are in the ice-cream industry and in the preservation of fruits, vegetables, and fish by the quick-freezing process. The rapid freezing and the extremely low temperature of the frozen product keep it in excellent condition for a long time. The superiority of dry ice when used to preserve food during shipment lies in its low temperature, light weight, and dryness as compared with ice made from water.

628. The Vacuum System of Refrigeration.—(a) The vacuum, steam-jet, or steam-ejector system of refrigeration may be applied where a moderate quantity of water at low temperature is required, where condensing water of suitable quality is readily available, and where the cost of steam is reasonable. One arrangement of a vacuum refrigerating system ³¹ is shown in Fig. 1029. The temperatures and vacuums carried vary in different installations, but those given in the diagram are typical. The warm water is sprayed into the *evaporator*, *D*, where a low absolute pressure is produced by the *primary steam ejector*; a small portion of this water immediately flashes into steam, absorbing the required heat of vaporization from the remaining water, which is chilled to the saturation temperature (40 deg. fahr.) corresponding to the pressure in the evaporator (0.25 in. Hg abs.). The chilled water is then pumped from the evaporator to the place of use. In order to maintain the high vacuum in the evaporator, the flash steam must be removed and compressed by the primary steam ejector, *ABC*, to a temperature sufficiently high to cause heat transfer from the steam to the available condensing water with consequent condensation of the steam in the *primary condenser*, *E*. If the condensing water is cold

³¹ Reproduced from "Cooling Water for Air Conditioning or Industrial Process Uses," by Walter L. Fleisher, in Heating, Piping and Air Conditioning, February, 1933, p. 77.

and plentiful, a high vacuum may be carried in the condenser thereby cutting down on the work required of the ejector; in some cases, however, the condensing water is scarce and must be cooled in ponds or towers and recirculated. Air that collects in the primary condenser is removed by using *secondary steam ejectors* to compress the air in two stages to atmospheric pressure, and the steam used by these ejectors is condensed in the *inter- and after-condenser*, *F*.

(b) A numerical example will further illustrate the principle of the vacuum system of refrigeration. Assume that one pound of water enters the evaporator at a temperature of 55 deg. fahr., while chilled water at a temperature of 40 deg. fahr. is required. The pressure that must be maintained in the evaporator is 0.25 in. Hg

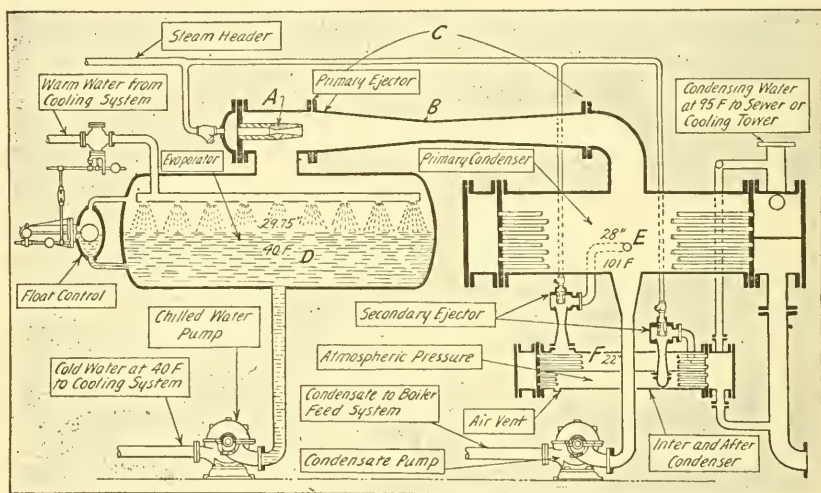


FIG. 1029.—A Vacuum Refrigeration System.

abs. Assuming no external transfer of heat in the evaporator, the flashing into saturated steam of 0.014 lb. of water would result in cooling the rest of the water (0.986 lb.) to 40 deg. fahr. Assume that the condensing water is available at such a temperature and in sufficient quantity to maintain a temperature of 101 deg. fahr. in the primary condenser; the condenser pressure would then be 2.0 in. Hg abs. The primary steam ejector must then compress the steam formed in the evaporator to this pressure and deliver it to the primary condenser. If saturated steam at a pressure of 100 lb. per sq. in. abs. were supplied the primary ejector, only about 0.5 lb. of this steam would be required to compress and deliver one pound of flash steam from the evaporator to the condenser under ideal conditions; actually, however, the efficiency of the ejector as a pump is such that about 3.5 lb. of steam per pound of flash vapor or about 0.05 lb. of steam per pound of chilled water would be required under the specified conditions. The amount of condensing water required would be about 2.4 lb. per lb. of chilled water, assuming an allowable temperature rise of 30 deg. fahr.

(c) The **principal application** of the vacuum refrigerating system is now, and probably will continue to be, in air conditioning. The low first cost, small space occupied, light foundations, lack of moving parts, and noiseless operation are advantages of importance in this field. In addition to air conditioning, there are other applications where the chilled water from the evaporator may be used directly, as in process condensers, coolers, and drinking-water systems.

629. The Ammonia Absorption Process.—(a) *In the ammonia absorption process of refrigeration*, heat is supplied directly to the refrigeration machine to accomplish the refrigerating effect. The condenser, expansion valve, and refrigerating coil are still a part of the equipment and have exactly the same functions as in the vapor compression process. The essential difference between the compression and absorption

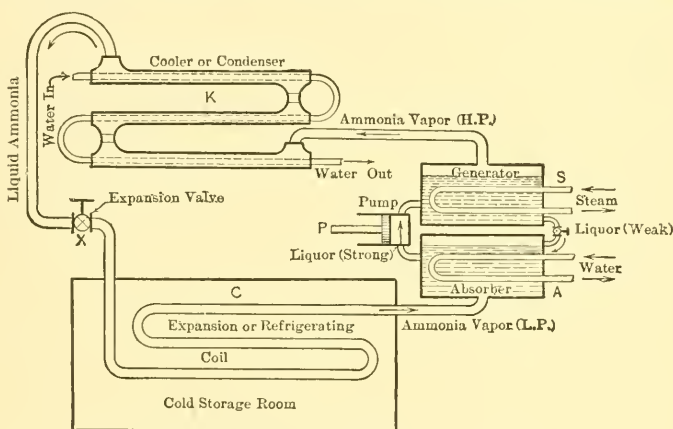


FIG. 1030.—Elementary Absorption System.

systems is the replacement of the compressor by three pieces of apparatus, viz., the absorber, the aqua ammonia pump, and the generator; hence the *elementary* absorption plant consists of the generator, condenser, expansion valve, refrigerating coil, absorber, and aqua ammonia pump, arranged with connecting piping as shown in Fig. 1030.

If the absorber, pump, and generator are to replace the compressor, the pressure and temperature of the ammonia vapor must be raised in passing through this equipment while flowing from the coil to the condenser. The possibility of doing this depends upon the properties of a solution of ammonia in water, or *aqua ammonia*. If the temperature of such a solution of a given concentration is increased at constant pressure, a temperature will be reached at which the water is unable to hold as much ammonia in solution as at the lower temperature, and as

a result ammonia vapor, and a small amount of water vapor, also, will escape from the solution. In other words, water can hold increasing amounts of ammonia in solution as either the pressure on the solution increases or as the temperature of the solution decreases.

(b) In the **generator**, heat is supplied to the aqua ammonia usually by circulating steam through coils, labeled *S* in Fig. 1030, immersed in the solution. The original concentration of the ammonia in the water may be chosen such that for the temperature of the solution as fixed by the temperature of the steam supplied, and the pressure on the solution as determined by the condenser action, some of the ammonia vapor will be driven off from the solution and will pass to the condenser. This leaves in the generator a weak aqua ammonia solution, and if the ammonia vapor is to be supplied steadily to the condenser, the generator must be constantly filled with a stronger solution furnished by the *absorber* and *pump*.

The **absorber**, shown at *A* in Fig. 1030, is arranged so that the weak solution, which has a greater density than the strong, may flow directly from the bottom of the generator to the absorber. The ammonia vapor, as it comes from the expansion coil at a low pressure and temperature, is brought into contact with this weak solution, which, because of its low temperature, is able to absorb the ammonia vapor, thereby increasing the ammonia concentration of the solution. Since heat is given off during this absorption of ammonia vapor, cold water must be circulated through the absorber in order to maintain a temperature which will be low enough to permit the absorption of the vapor by the weak solution. The **pump** then delivers the strong solution to the generator, and the cycle of the aqua ammonia is thereby completed.

(c) *Certain additional apparatus is commonly used to make the economy of operation better than that of the elementary plant.* One such device is the *exchanger* or *economizer*. Since the weak solution leaves the generator at a high temperature, and since the strong solution leaves the absorber at a low temperature, the solutions are made to flow in opposite directions through the exchanger, and an exchange or transfer of heat is effected which improves the performance.

The ammonia vapor leaving the generator carries some water vapor or steam. The amount of water vapor in the mixture may be from 5 to 13 per cent by weight. The condensation of this water vapor would result in an undesirable accumulation of water in the condenser or in the refrigerating coils. To remove this water, either an *analyzer* or a *rectifier*, or both, may be used.

The *analyzer*, which is sometimes omitted in present designs, con-

sists of a series of trays above the generator. The vapors leaving the generator pass up over these trays, while the strong solution from the exchanger flows down over the trays on its way to the generator. There is a transfer of heat between the two fluids, and some of the water vapor is carried back to the generator.

To reduce still further the water content of the vapor, a *rectifier* may be placed between the analyzer and condenser, or it may be used alone between the generator and condenser. The rectifier is simply a water-cooled preliminary condenser, in which the vapor is cooled sufficiently to condense most of the water vapor. The temperature to which the vapor is cooled is usually about 20 deg. fahr. higher than the temperature of saturated anhydrous ammonia vapor at condenser pressure. The condensate consists largely of water, although there will be some ammonia condensed to maintain an equilibrium with the vapors; this condensate is returned to the generator.

Economy in the use of cooling water is secured by using the same water for several cooling operations. For example, the coldest water may first be forced through the condenser; then it may flow in succession through the absorber, exchanger, and rectifier. Such a sequence of use is possible, because each device along the line of flow will perform satisfactorily with water at successively higher temperatures.

(d) The **properties of aqua ammonia solutions** cannot be fully given in this text; but even for an elementary study of the absorption system of refrigeration certain essentials need to be explained. The temperature at which an aqua ammonia solution of a given concentration will boil has been studied by Hilde Mollier, and the following equation fits the experimental data over the range of concentrations used in the ammonia absorption process:

$$\frac{T_s}{T_b} = 0.466x + 0.656 \quad . \quad . \quad . \quad . \quad . \quad . \quad (709)$$

where T_s = the saturation temperature of anhydrous ammonia vapor corresponding to the pressure of the solution, in deg. fahr. abs. (See Table E, Appendix.)

T_b = the boiling temperature of the aqua ammonia, in deg. fahr. abs.

x = the fraction, by weight, of liquid ammonia in the solution.

A few values of the boiling temperatures of aqua ammonia are also given in Table C.

TABLE C

BOILING TEMPERATURES OF AQUA AMMONIA IN DEG. FAHR.

Fraction of NH ₃ by Weight *	Absolute Pressure, Pounds per Square Inch										
	2	4	10	20	30	40	60	80	100	150	200
0.00	126.1	153.0	193.2	228.0	250.3	267.2	292.7	312.0	327.8	358.4	381.8
0.02			169.0	206.5	231.1	249.4	277.9	299.0	316.2	350.9	377.7
0.04			161.0	197.5	221.9	240.3	267.7	288.6	305.6	342.0	365.7
0.06			153.0	188.8	212.4	229.5	258.0	278.5	295.3	337.9	354.2
0.08			143.5	180.1	203.7	221.4	248.2	268.5	285.1	317.3	343.2
0.10			137.4	171.1	194.6	212.3	239.6	259.5	275.8	307.4	331.6
0.12			129.0	163.1	185.5	203.3	229.2	249.4	265.6	297.0	320.7
0.14			120.5	155.5	177.6	195.2	220.8	240.5	256.1	287.3	311.1
0.16			113.0	147.4	169.4	187.1	212.0	231.6	247.3	277.5	301.3
0.18			107.5	139.7	161.2	178.3	203.1	222.6	238.0	268.0	291.7
0.20			99.0	132.0	153.8	170.2	195.4	213.8	229.1	258.6	282.7
0.22			92.0	124.6	146.0	162.7	187.0	205.6	220.6	250.0	273.0
0.24			86.0	117.0	139.2	155.2	180.0	198.4	213.0	241.9	264.5
0.26			79.2	110.6	131.9	148.2	172.1	190.7	205.1	233.7	255.8
0.28			73.4	103.6	125.0	140.6	165.2	182.8	197.1	225.5	247.5
0.30			66.5	97.0	117.8	133.5	157.0	174.9	189.4	217.3	239.3
0.40			37.0	66.0	86.0	101.0	123.4	140.0	153.4	180.1	200.1
0.50			11.0	38.9	56.8	70.9	92.6	108.3	121.0	146.4	166.0
0.60	-52.0	-30.5	-1.5	26.0	43.5	57.5	78.0	93.5	106.0	130.5	149.0
0.70	-65.0	-42.5	-14.0	12.5	30.0	43.0	63.0	78.0	90.5	114.5	132.5
0.80	-74.5	-53.0	-24.5	1.0	18.0	31.0	51.0	65.5	77.0	100.5	118.5
0.82	-77.0	-54.5	-26.5	-1.0	16.0	29.0	48.5	63.0	75.0	98.0	116.0
0.84	-79.5	-55.0	-28.5	-3.0	14.0	27.0	46.0	61.0	72.5	95.5	113.5
0.86	-80.5	-57.5	-30.5	-5.0	12.5	25.0	44.0	58.0	70.0	93.0	110.5
0.88	-81.5	-59.5	-32.5	-6.5	10.0	23.0	42.0	56.5	68.0	91.0	108.5
0.90	-83.0	-62.0	-34.5	-8.0	8.0	21.0	40.0	54.5	66.0	88.5	106.0
0.92	-84.0	-63.5	-35.5	-10.5	6.5	19.0	37.5	52.0	64.0	86.5	104.0
0.94	-85.0	-65.0	-37.0	-12.0	4.5	17.5	35.1	50.0	62.0	84.5	101.5
0.96	-85.5	-66.0	-38.5	-13.5	2.5	15.5	33.5	48.5	60.0	82.5	99.5
0.98	-86.0	-67.0	-40.0	-14.5	1.0	13.5	31.5	46.5	58.0	80.5	97.5
1.00	-90.0	-68.5	-41.3	-16.6	-0.6	11.7	30.2	44.4	56.0	78.8	96.3

* The values in this column are commonly called the *concentration*, x , or better the *weight concentration* of the solution. Hence $x = \frac{z}{y+z}$, where the solution consists of z lb. of ammonia and y lb. of water.

The *molar concentration*, x_m , is also often used, and since the molecular weights of NH₃ and H₂O are approximately 17 and 18, respectively, it follows that $x_m = \frac{z \cdot 17}{y \cdot 18 + z \cdot 17} = \frac{z}{0.944 y + z}$.

See Bull. 146, Univ. of Ill., "The Total and Partial Vapor Pressures of Aqueous Ammonia Solutions," by Thomas A. Wilson.

As a **numerical illustration** of the use of Eq. (709), assume that exhaust steam at a pressure of 30 lb. per sq. in. abs. is available for heating the generator, and that the available cooling water maintains a condenser-pressure of 150 lb. per sq. in. abs., which is also the pressure on the solution in the generator. The temperature which

These quantities of energy may be calculated on the unit time basis or on the basis of the circulation of one pound of vapor refrigerant.

The analysis, however, is far from simple. The refrigerant leaving the generator is not anhydrous ammonia but a mixture of ammonia vapor and water vapor. Although most of the water vapor is removed in the rectifier, a small amount passes through the rest of the cycle. It is necessary, then, to have information concerning the constituents of the vapors above aqua ammonia solutions at different pressures and temperatures;³² for example, it may be shown that, for the conditions of the numerical illustration of (d), about 7.5 per cent, by weight, of the vapor driven off from the aqua ammonia solution in the generator would be water vapor.

Also, the heat content of the aqua ammonia solution must be known. This is the sum of the weight of the water times its heat content and the weight of the ammonia liquid times its heat content.

The heat of solution, or the amount of heat generated when the ammonia concentration of an aqua ammonia solution is increased, as in the absorber, must be known. The equation commonly used for the heat of solution per pound of the liquid ammonia absorbed is derived from the work of Mollier and follows:

$$q_s = 345(1 - x) - 400x^2 \quad . \quad . \quad . \quad . \quad (712)$$

where q_s = the heat of solution, in B.t.u. per lb. of ammonia absorbed.
 x = the average fraction by weight of ammonia in solution
 $= (1/2)(x_1 + x_2)$.

With the necessary properties of the aqua ammonia solutions and the mixtures of ammonia and water vapor known, the energy balance for the absorption system may be further analyzed. In general, the heat supplied the *generator* is *usefully* employed: (1) to raise the temperature of the entering strong solution to that of the leaving weak solution; (2) to separate the ammonia from the solution (the heat of solution); (3) to vaporize and superheat the ammonia; and (4) to vaporize and superheat the water that leaves the generator with the ammonia vapor. In the *absorber*, the quantity of heat that is *usefully* removed consists of: (1) the heat abstracted from the entering weak solution in cooling it to the exit temperature of the strong solution; (2) the heat generated during the absorption of liquid ammonia (the heat of solution); and (3) the heat abstracted from the entering refrigerant vapor in its condensation and delivery at the exit temperature of the strong solution.

(g) The **coefficient of performance** of an absorption machine is the

³² See "The Total and Partial Vapor Pressures of Aqueous Ammonia Solutions," by T. A. Wilson, Bull. 146, Univ. of Ill. Engg. Expt. Sta.

ratio of the heat abstracted from the cold body to the energy required to operate it; and the latter energy consists of that supplied in the form of heat to the generator plus the mechanical energy required to operate the pump. If the pump is steam driven, its exhaust is used to supply a small part of the steam required by the generator. Usually the coefficient of performance of the absorption machine is not given; but instead its performance is expressed in terms of the amount of steam required under specific conditions to produce a ton of refrigeration.

(h) The **application of the absorption system** is best suited to those fields requiring very low temperatures, say below 0 deg. fahr., and where exhaust steam is available. Under these conditions the cost of operation may become very low because the exhaust steam might otherwise be wasted. In general, however, the compression system is preferred, and the absorption system is not used in any large plants, say those having capacities greater than 200 tons per day. For capacities from 50 to 100 tons per day and a low temperature of the refrigerator, the absorption system and the compression system may both give about the same performance; but the absorption system has a much higher first cost and generally requires more attention from the operator. For very small capacities, the absorption system is also sometimes used, as described in Sect. 632, for household machines.

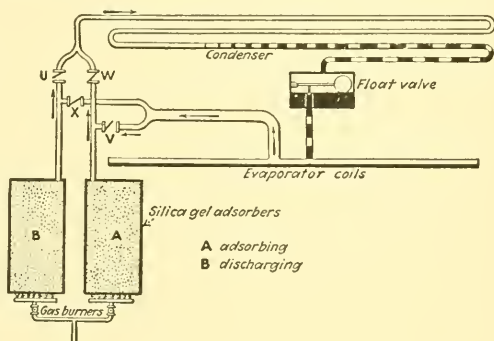


FIG. 1031.—A Silica Gel Adsorption System of Refrigeration.

630. The Adsorption System of Refrigeration.—(a) A solid material, such as *silica gel*, may be used to adsorb a refrigerant vapor which may later be driven from the solid by heating. Silica gel, or silicon dioxide, resembles clear quartz sand and is extremely porous, each particle exposing a large surface that will absorb relatively great quantities of vapor. Sulphur dioxide is the refrigerant commonly used with silica gel, and a typical adsorption system for continuous refrigeration is shown in Fig. 1031. The silica gel container, *B*, releases sulphur dioxide vapor, previously adsorbed, as heat is supplied; the pressure rises, closes the valve *X* and opens valve *U*. The sulphur dioxide vapor then passes to the condenser, is condensed, and flows through the float chamber to the evaporating coils. The vapor formed in the coils then

passes through *V* to container *A*, which is adsorbing gel at the time. The functions of the containers *A* and *B* are then reversed; the burner under *B* is shut off automatically, and as the pressure drops the valve *U* closes. When the pressure in *B* drops below that in the evaporator, the valve *X* opens, and the silica gel in *B* starts adsorbing the refrigerant vapor, while the heat supplied by the burner under *A* drives off sulphur dioxide vapor through *W* to the condenser.

(b) The **principal application** of the silica gel adsorption system has been in the cooling of refrigerator cars. One fuel used is propane, carried in tanks under the car. The condenser coils are placed on the roof of the car and are air cooled.³³

631. Applications of Refrigeration.—(a) The chief applications of mechanical refrigeration are found in the following fields: ice making, cold storage, certain manufacturing industries, transportation of food products, air conditioning, and household refrigeration. These cover an enormous range and affect the lives of millions of people; much of our present mode of living is directly dependent upon the proper functioning of our refrigerating equipment.

(b) One of the first applications of mechanical refrigeration was the *manufacture of ice*. At the present time, there are two systems of ice making in general use, the *can* system and the *plate* system.

With the *can* system, the water to be frozen is in metal cans, which are immersed in large tanks containing brine which circulates around the cans after being cooled by the refrigerant. The water is kept agitated by compressed air so that it freezes into clear cakes.

With the *plate* system, the water to be frozen surrounds large metal plates, which contain coils or cells filled with the refrigerant or with cold brine. With this system, the ice freezes into single cakes weighing several tons.

(c) Since temperatures below 40 deg. fahr. materially impede the growth of the micro-organisms which cause decay and spoilage, mechanical refrigeration is used very extensively for the maintenance of these temperatures in the *cold storage* and *transportation of foodstuffs*.

To prevent damage from moths and beetles, furs and fabrics may be stored during the summer months at temperatures around 40 deg. fahr.

(d) The application of refrigeration to *air conditioning* has been treated in Chapter L, and a brief discussion of household refrigerators is given in the following section.

632. Household Refrigerators.—(a) In recent years many small refrigerating units have been installed in households. These units are

³³ See Power, Vol. 69, 1929, p. 50.

usually of the compression type, but in some the absorption system is used. In all cases, however, the units are compact and are specially designed to be free from trouble and thus to require a minimum amount of attention. Some of the compressors are of the rotary type whereas others are of the reciprocating type. The compressors are driven by electric motors, and the expansion valves are entirely under automatic control. Only a few of the different makes can be considered here.

(b) The **Norge** uses a rotary compressor, or "rollator," as shown in Fig. 1032. The refrigerant used is sulphur dioxide, and the compressing element is a roller moving in a closed cylinder and mounted on an eccentric. The roller does not revolve at shaft speed but rolls slowly around the eccentric portion of the shaft. The inlet and discharge passages are always separated by the blade that maintains contact with the roller surface. The shaft, blade, and roller are submerged in oil, and there is an oil film between all sliding surfaces. The point of

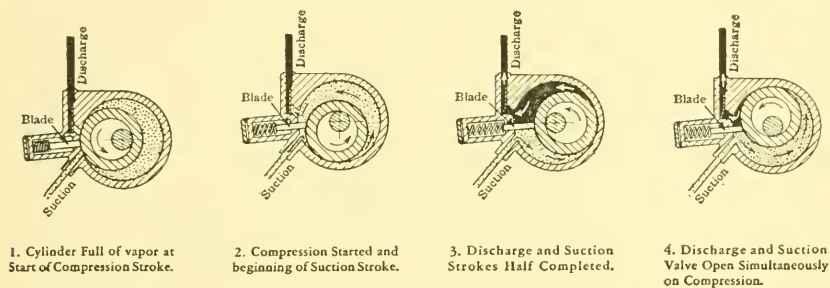


FIG. 1032.—The Norge Rollator.

minimum clearance moves around the inside of the cylinder as indicated, and an oil film at that point maintains the seal between the high- and low-pressure sides of the roller. The rest of the unit consists of a motor driving the compressor, an air-cooled fin-tube condenser, a receiver, and an evaporator with float-valve control, all quite similar in fundamental principle to the refrigerating machine shown in Fig. 1034.

(c) Some of the details of the **General Electric** household refrigerating machine, Model CA-2, are given in Fig. 1033, in which the parts are shown separated from each other for clarity. This machine is of monitor-top design, hermetically sealed, and when completely assembled, forms a single unit that may readily be placed on or removed from the cabinet. The refrigerant is methyl formate, and the compressor is of the rotary type. The oscillator of the compressor is operated by an eccentric on the motor shaft directly below the induction motor. The top end plate and motor frame are combined in one piece which is

bolted to the base and is spring-supported at three points to prevent noise and vibration. The oscillator is keyed to the cylinder by the

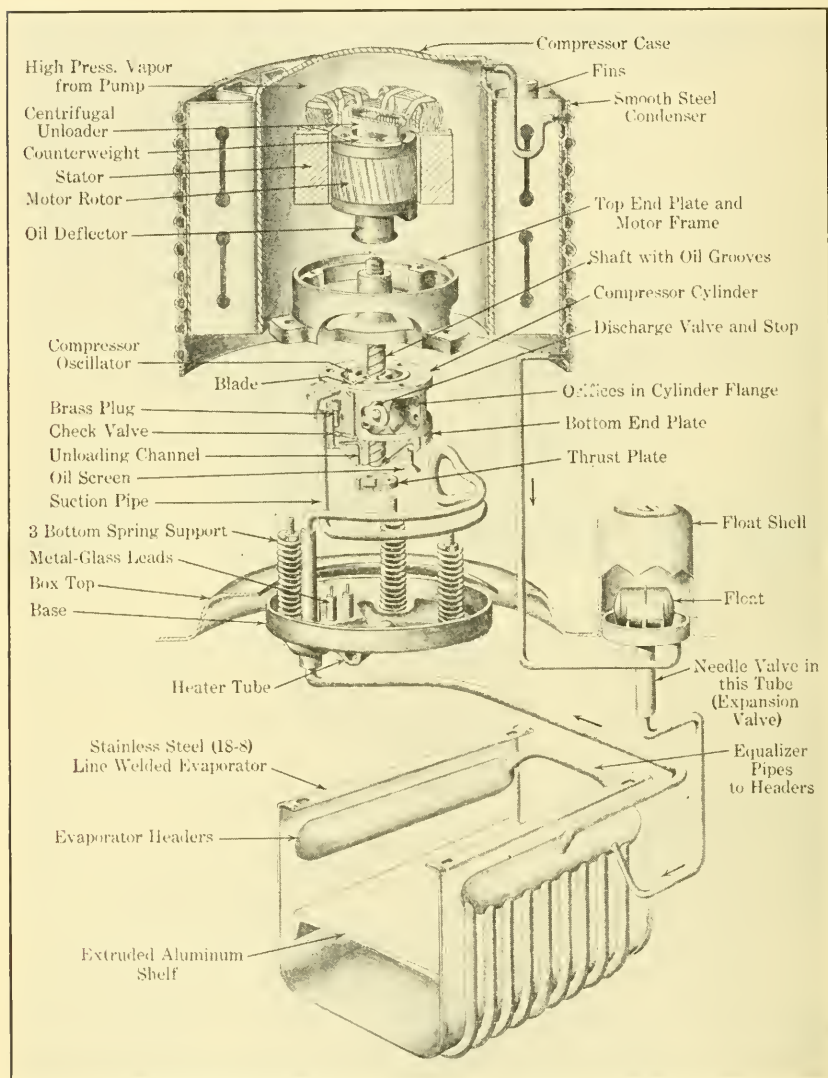


FIG. 1033.—Elements of the General Electric Machine.

blade as shown, and as the eccentric rotates the oscillator does not rotate but oscillates, so that the point of minimum clearance between the oscillator and the cylinder rotates around the inside of the cylinder

wall. On the suction side of the oscillator, low-pressure vapor is drawn into the compressor through the suction pipe that connects with the evaporator, or chilling unit. This vapor is compressed and expelled through a discharge valve of the disc type, thence flows through orifices in the cylinder flange, that serve as acoustic mufflers, into the inside of the compressor case. The refrigerant then passes through the condenser where it condenses by giving up heat to the air circulating around the outside of the condenser fins. The liquid refrigerant collects in the float chamber until there is a sufficient quantity to lift the float enough to allow the liquid to expand through the expansion valve into the evaporator. In the evaporator, suitably designed nozzles force a circulation of the refrigerant, and, as heat is removed from the contents of the chilling unit and from the air in contact with the evaporator walls, the refrigerant vaporizes and then returns to the compressor.

To reduce the starting torque, the compressor is unloaded automatically, and the lubricating oil is kept warm by a heater. In order to unload the compressor, the pressures on the suction and discharge sides must be equalized. This is accomplished by the use of a centrifugal unloader, consisting of a plunger and spring in a housing on the top of the shaft. During normal running operation, the plunger closes an opening in this housing that connects with a hole drilled down through the shaft, through an unloading channel in the thrust plate and up into the check valve chamber in the compressor cylinder; but as the motor slows down, the plunger moves toward the center of the shaft, uncovering this passage and allowing high-pressure vapor to pass into the suction side of the compressor through the open check valve. At the time of stopping, the check valve closes, and prevents the passage of the high-pressure vapor into the suction line and evaporator. The viscosity of the oil is reduced by a heating wire inserted through the heater tube in the base; this heater also drives off any refrigerant held by the oil in the base. The oil is pumped from the base through the screen and grooves in the shaft, under pressure, to all moving parts.

The control, not shown, includes a manual switch for turning the machine on or off, an adjustable automatic mechanism for regulating the cabinet temperature by means of a control of motor running time, a device for protecting the motor from abnormal loads, and a semi-automatic arrangement for defrosting the chilling unit.

With the motor mounted inside of the compressor case, the necessity of using a stuffing box is eliminated, but the motor losses, which are dissipated in the form of heat, increase the cooling load to be handled.

(d) The **Frigidaire** refrigerating machine is shown in Fig. 1034. For household use, the motor, compressor, condenser, and receiver are

mounted in the base of the refrigerator cabinet beneath the refrigerated space. The evaporator is the only element in the cooled space. The refrigerant used with the machine shown in Fig. 1034 is sulphur dioxide, but dichlorodifluoromethane (F-12) is also used in later equipment of the same type. The refrigerant vapor enters the cylinder of the single-acting compressor through the valve in the piston head, is compressed and discharged through the flapper or reed-type check valve into the condenser. In the household unit, the condenser is of the finned-radiator type, and the refrigerant is condensed by the transfer of heat

to air passing over the outside of the condenser surface. The liquid refrigerant then flows into the receiver. If the **flooded system** is used, the liquid refrigerant flows from the receiver through the expansion valve, which is under float control, into the evaporator. As the liquid refrigerant absorbs heat from the cabinet and evaporates in the finned cooling coil the float drops and opens the needle of the expansion valve, thus permitting additional liquid refrigerant to expand into the cooling

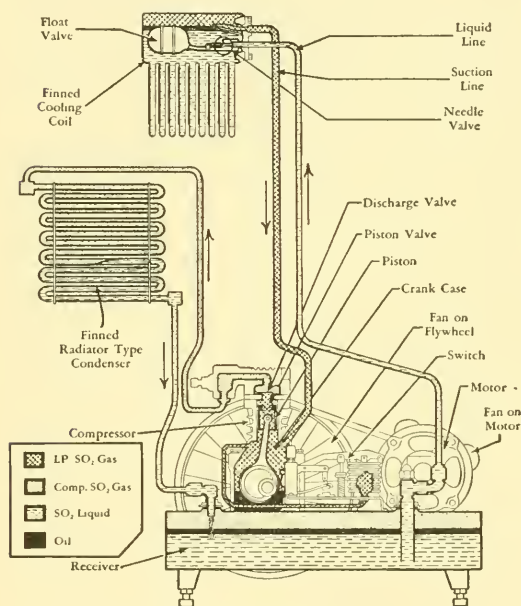


FIG. 1034.—The Frigidaire Flooded System.

coil. The vapor formed in the cooling coil returns through the suction line to the crank case of the compressor.

If the **dry expansion** system is used, the only difference is in the refrigerant control valve and evaporator. In the expansion system shown in Fig. 1035 the admission of refrigerant to the evaporator is controlled by the thermostatic expansion valve. The reception of heat by the refrigerant in the evaporator raises the pressure on the switch bellows and starts the motor. At the same time, the rise in temperature of the volatile liquid in the thermostat bulb raises its pressure, and this rise in pressure is transmitted to a bellows in the expansion valve

cap. The pumping action of the compressor also lowers the pressure on the evaporator side of the expansion valve. These simultaneous pressure changes produce a pressure differential sufficient to open the needle valve and allow liquid refrigerant to expand into the evaporator. As the compressor continues to operate, the temperatures of the evaporator and of the thermostat bulb drop, the pressure of the volatile liquid on the bellows in the valve cap drops, and the needle valve closes. The reduction in refrigerant pressure then causes the switch bellows to contract and shut off the motor.

The motor drives the compressor by means of one or two V-type belts. The flywheel on the compressor is provided with fan blades that force the air over the surface of the condenser and out at the back of the cabinet. With the motor and compressor outside of the refrigerated space, the motor losses that are dissipated in the form of heat do not directly increase the cooling load, but at the same time, a stuffing box of good design is necessary to prevent the leakage of refrigerant into the room air.

(e) The Servel **Electrolux** gas refrigerator is an application of the absorption system to a household refrigerator. The refrigerant is ammonia, and the elements of the system are the gas burner, generator, rectifier, condenser, gas heat exchanger, evaporator, absorber, and liquid heat exchanger, arranged as shown in Fig. 1036. The condenser and absorber shown in this figure are water cooled, but later units are air cooled. The principal difference between this household refrigerator and the absorption units described previously is that the use of a pump

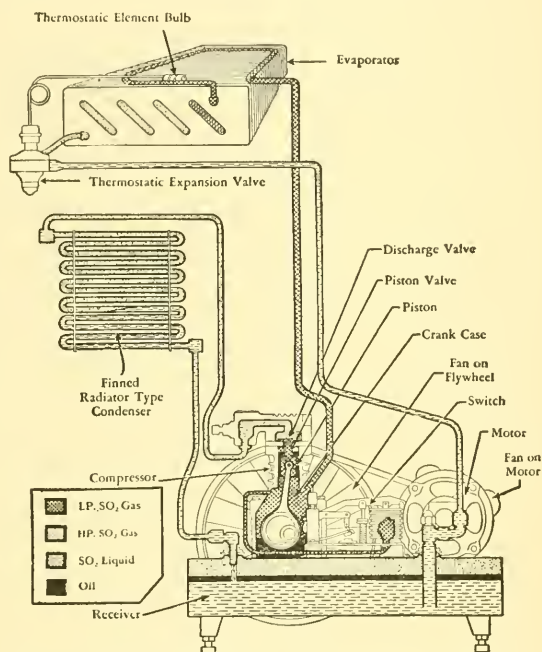


FIG. 1035.—The Frigidaire Dry Expansion System.

to transfer the strong aqua ammonia from the absorber to the generator is ingeniously avoided, so that, with air cooling, the only necessary attachment is to the gas main.

During operation of the **water-cooled unit**, slugs of aqua ammonia liquid and vapor rise in the small vertical pipe from the bottom to the top of the generator as heat is supplied by the gas burner of the Bunsen type. The vapor, liberated in the upper chamber by the absorption of heat, leaves the generator and flows to the rectifier which is jacketed with liquid ammonia supplied through the pipe *C*. Water vapor is

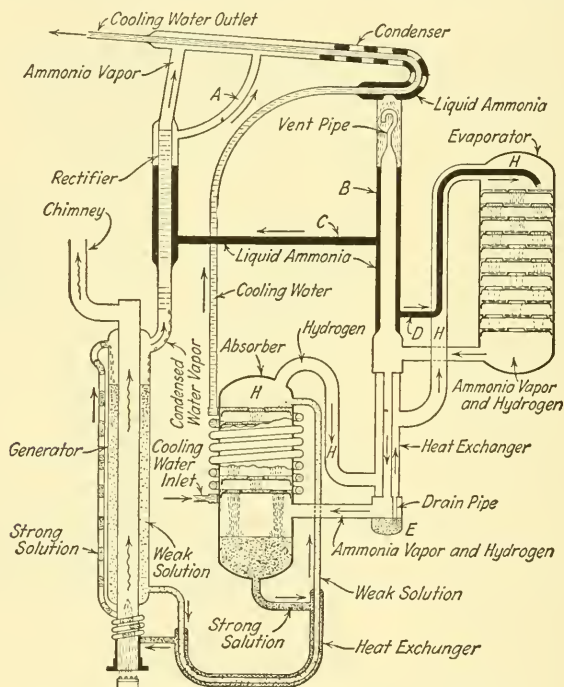


FIG. 1036.—Elements of Water-cooled Electrolux.

condensed in the rectifier and is then returned to the generator. The ammonia vapor then flows to the condenser, wherein it condenses as heat is removed by the cooling water (or by air). The closed system, made up of the absorber (which is water sealed), the gas heat exchanger, and the evaporator, is charged with hydrogen gas under a partial pressure of about 150 lb. per sq. in. The ammonia liquid enters this closed system from the condenser, flowing through the pipe *D* to the evaporator. Since the pressure of the ammonia liquid is about 180 lb. per sq. in., while the partial pressure of the hydrogen (*H*) in the evaporator is 150,

the ammonia will expand irreversibly to produce a total pressure of 180 lb. per sq. in., or an ammonia partial pressure of 30, in the evaporator. The density of the mixture of ammonia vapor and hydrogen gas is greater in the evaporator than in the absorber, because of different temperatures and the different proportions of the two constituents; a circulation is therefore set up, with a downward flow in the evaporator and an upward flow in the absorber. The mixture of ammonia vapor and hydrogen gas passes through the gas heat exchanger to the absorber, in which the weak solution of aqua ammonia absorbs ammonia vapor, since the partial pressure of this vapor is only 30 lb. per sq. in. The strong solution, so formed, collects in the bottom of the absorber, from which it returns through the liquid heat exchanger to the lower coils of the generator, while the weak solution flows in the opposite direction from the generator, through the exchanger, to the absorber. Since the hydrogen is not absorbed by the liquid in the absorber, the pressure maintained on this liquid is sufficient to return the aqua ammonia to the generator without a pump.

The **air-cooled** Electrolux avoids the necessity of connecting the unit to any water system; this is accomplished (1) by substituting an *ammonia condenser cooled by air* instead of water, (2) by substituting an *absorber cooled by methyl chloride* instead of water, and (3) by adding a *methyl chloride condenser* cooled by air. The arrangement of the various parts of the unit is shown in Fig. 1037. The operation of the unit, as given by the manufacturer's copyrighted explanation and reproduced by permission, is as follows:

The Electrolux unit is charged with a small quantity of aqua ammonia (distilled water and ammonia) and hydrogen. The charge distributes itself naturally in the unit, the liquid seeking the lowest levels and the hydrogen filling the remaining space.

With the application of heat at the Generator (1), ammonia vapor is driven off from the strong solution (aqua ammonia) and together with small quantities of the strong solution is raised through the Pump Tube (2) to the Weak Liquid Stand Pipe (16). Ammonia vapor with traces of water vapor is driven off in the Generator (1) leaving the aqua ammonia solution comparatively weak in ammonia (weak solution). The hot ammonia vapor passes from the Generator (1) through the Submerged Analyzer (3) High Temperature Rectifier (5) and Rectifier (6) where the small amount of water vapor entrained is condensed and drains to the Generator (1).

The hot ammonia vapor reaches the Lower Ammonia Condenser (7) where part of it is liquefied by cooling. The liquid ammonia maintains a level in the Rectifier (6) causing the liquid ammonia to flow into the Freezing Evaporator Coil (8). The remaining part of the ammonia vapor is liquefied in the Upper Ammonia Condenser (7A) and the resultant liquid flows to the Box Cooling Evaporator Coil (9). In the evaporator the ammonia liquid evaporates with the resulting absorption of heat.

In a cold room the greater part of the ammonia vapor liquefies in the Lower Ammonia Condenser (7) and is available for freezing cubes by evaporation in the Freezing Evaporator Coil (8). Since very little liquid flows to the Box Cooling

Evaporator Coil (9) it is possible to obtain fast freezing without excessive cooling of the box.

As room temperature or refrigeration load demand increases the thermostat functions to increase the generator heat input and raises a greater amount of ammonia vapor to the condenser. This additional vapor is condensed in the Upper Ammonia Condenser (7A) and flows to the Box Cooling Evaporator Coil (9). This arrangement tends to maintain a more uniform box temperature with varying room temperature without the necessity of changing the temperature control setting.

An atmosphere of hydrogen gas continually sweeping the surface of liquid ammonia in the evaporator coils keeps removing the ammonia vapor and causes con-

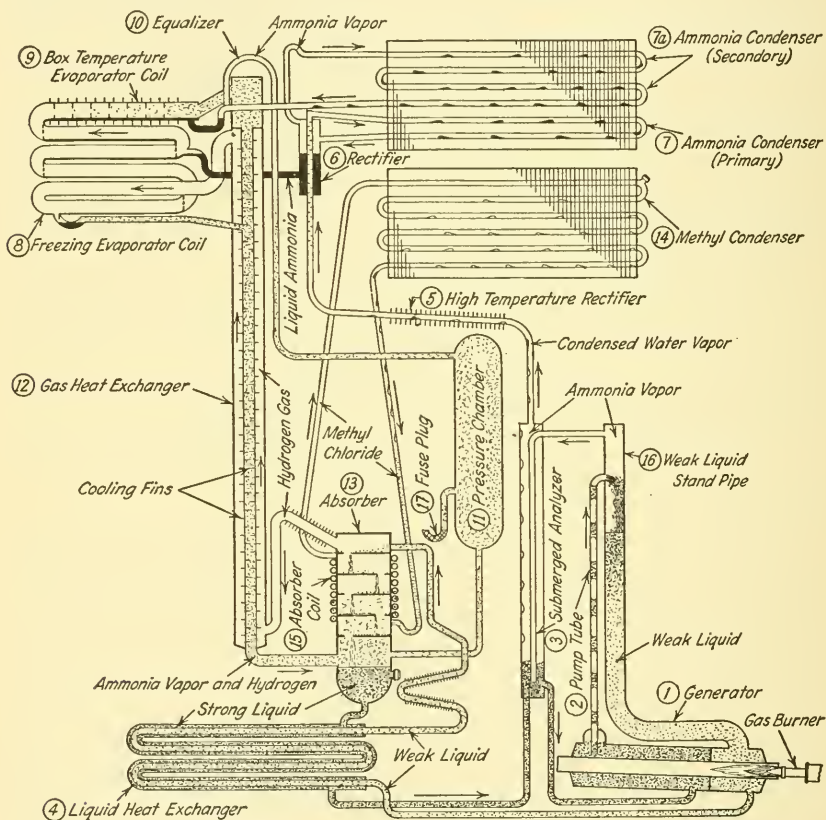


FIG. 1037.—Elements of an Air-cooled Electrolux.

tinued evaporation. The ammonia vapor thus formed in the evaporator mixes with hydrogen gas and the mixture is made to flow upwardly through the evaporator. This permits the location of the Box Cooling Evaporator Coil (9) in its most effective position; namely, at the top; and at the same time, the hydrogen gas which is saturated with ammonia passing upward from the Freezing Evaporator Coil (8) is enabled to pick up more ammonia at the higher temperature prevailing in the Box Cooling Evaporator Coil (9). Still another advantage results from the pre-cooling

of the liquid ammonia coming from the Ammonia Condenser—Upper Part (7A) before it enters the Freezing Evaporator Coil (8). These features greatly increase the capacity of the unit.

In case all the ammonia vapor is not condensed in Upper Ammonia Condenser (7A) on account of high room temperature, the vapor passes through the Equalizer (10) to the Pressure Chamber (11) which contains a quantity of reserve hydrogen. This results in displacement and transfer of reserve hydrogen into the Absorber (13) thereby increasing the pressure in the system to the point where all the ammonia is liquefied in the Lower Ammonia Condenser (7) and Upper Ammonia Condenser (7A). When the room temperature becomes lower, the reserve hydrogen returns to the Pressure Chamber (11). This automatic pressure change permits very efficient operation of the unit under normal room temperature conditions as it is not necessary to initially charge the unit with a high pressure to enable satisfactory operation under extreme conditions.

In the Gas Heat Exchanger (12) cool heavy gas from the evaporator comes in thermal contact with warm light gas from the Absorber (13) effecting a heat exchange and increasing the unit efficiency. The long column of heavy gas rich in ammonia in the center of the Gas Heat Exchanger (12) readily overbalances the short column of heavy gas in the evaporator, thereby causing the desired upward flow in the evaporator.

A flow of weak solution being returned from the Generator (1) through the Liquid Heat Exchanger (4) contacts the ammonia and hydrogen gas mixture entering the Absorber (13) and the ammonia is dissolved. The hydrogen returns to the evaporator.

The heat which is liberated by absorption of ammonia in the Absorber (13) is carried away by the vaporization of a small quantity of methyl chloride in the Absorber Coil (15). This vapor liquefies in the Methyl Chloride Condenser (14) thereby dissipating the heat. The liquefied methyl chloride returns by gravity to the Absorber Coil (15).

From the Absorber (13) the strong solution is returned by gravity to the Generator (1), passing through the Liquid Heat Exchanger (4) where a heat transfer is accomplished because of the difference in temperature of the weak and strong solutions. After passing it then returns to the Generator (1) through bottom of submerged Analyzer (3) where part of the heat in the vapors leaving the Generator (1) is recovered by serving the useful purpose of driving some ammonia from the strong solution. This results in a further increase in efficiency. Continued refrigeration is merely a repetition of this cycle.

(f) The **performance** of household refrigerating machines is naturally far below that of the large industrial ones. Not only are the household units relatively very small, but they are required to be simple and easy to install. This means that they nearly always use air instead of water as the cooling medium for the condenser, and thus the condenser temperature is relatively high.

An indication of what the small units will do is given by the test data ³⁴ of the Bureau of Standards on a machine using CH_3Cl as a refrigerant and driven by a $\frac{1}{4}$ -hp. motor. Some of these data are given in Table CI.

³⁴ See "The Evaluation of the Fractional Ton Refrigerating Machine," by J. B. Churchill, Refrigerating Engineering, March 1928, p. 67.

TABLE CI
TEST OF A HOUSEHOLD REFRIGERATOR USING METHYL CHLORIDE
By the Bureau of Standards

Temperatures, in Deg. Fahr.					Pressures Lb. per Sq. In. Absolute		Energy Re- quired in Kw-hr. per Ton	Coefficient of Performance				Overall Efficiency Referred to Ideal Vapor Cycle
Room	Com- pressor Dis- charge	Condenser Tube, Outside			Refrigerant			Overall	Ex- cluding Motor	Carnot* CH ₂ Cl	Ideal Vapor Cycle CH ₂ Cl	
		1st Coil	10th Coil	19th Coil	After Expan- sion Valve	To Com- pressor	High					Low
76	220	146	107	101	12.7	89.2	75	1.14	1.81	5.17	4.61	0.25
73	213	143	103	99	12.9	84.7	68	1.24	1.97	5.43	4.88	0.25
73	211	135	98	94	13.9	84.4	72	1.17	1.86	5.11	4.55	0.26
118	297	176	131	126	6.6	126.9	117	0.72	1.14	3.98	3.38	0.22
81	228	155	110	105	14.2	95.2	65	1.30	2.06	4.90	4.31	0.30
76	221	149	106	102	13.8	90.7	63	1.33	2.11	4.96	4.39	0.30
76	209	142	102	97	14.9	84.0	59	1.43	2.27	5.38	4.81	0.30

* The coefficient of performance of the Carnot cycle is here based on a low temperature to agree with that of the refrigerant measured after the expansion valve, and with a high temperature to agree with the saturation temperature of the refrigerant corresponding to the measured value of the high pressure.

APPENDIX.

USE OF COMMON LOGARITHMS FOR SPECIAL CASES.

CASE I. TO DETERMINE THE n TH POWER OF A NUMBER
LESS THAN UNITY.

Example: Find $0.5^{1.55}$ by logs.

In general $\log_{10} V^n = n \log_{10} V$; and in this case $V = 0.5$ and $n = 1.55$.

From the tables $\log_{10} 0.5 = 9.6990 - 10$,

Then, $1.55 \log_{10} 0.5 = 1.55 (9.6990 - 10) = 15.033450 - 15.5$

Subtract 5.5 to reduce the negative part of
the characteristic to 10,

$$\begin{array}{r} \phantom{\text{Log. of answer}} \\ \phantom{\text{Corresponding number}} \end{array} \begin{array}{r} 5.5 - 5.5 \\ \hline 9.533450 - 10.0 \\ 0.342 = 0.5^{1.55} \end{array}$$

(Note that a fraction raised to a power greater than unity gives a result less than the original fraction.)

CASE II. TO DETERMINE THE n TH ROOT OF A FRACTION.

Example: Given $V^{1.5} = 0.75$; Find V . Evidently, —

$$\log_{10} V = \log_{10} (\sqrt[1.5]{0.75}) = \log_{10} \left(0.75^{\frac{1}{1.5}} \right) = (\log_{10} 0.75) \div 1.5,$$

which is in the general form of $\log_{10} V = (\log_{10} C) \div n$,

where $C = 0.75$ and $n = 1.5$.

From the tables $\log_{10} 0.75 = 9.8751 - 10$.

$$\text{Then} \quad \frac{(\log 0.75)}{1.5} = \frac{(9.8751 - 10)}{1.5} = 6.5833 - 6.666.$$

Add 3.334 to raise the negative part
of the characteristic to 10,

$$\begin{array}{r} 3.3340 - 3.334 \\ \hline \text{Log. of } V = 9.9173 - 10.000 \end{array}$$

The corresponding number is 0.8266 which is $\sqrt[1.5]{0.75}$.

TABLE A. — COMMON LOGARITHMS (Log_{10}).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
0	0	0000	3010	4771	6021	6990	7782	8451	9031	9542	
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	42
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	38
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	35
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	32
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	30
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	28
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	21
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	20
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	18
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	11
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	8
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	8

$$e = 2.71828$$

TABLE A. (*Concluded*). — COMMON LOGARITHMS (Log_{10}).

No.	0	1	2	3	4	5	6	7	8	9	Diff.
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	7
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	6
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	5
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	4

Napierian $\log_e = 2.302 \log_{10}$.

TABLE B.—NAPIERIAN LOGARITHMS ($\log_e$).
(Also Called Hyperbolic or Natural Logarithms)

No.	0	1	2	3	4	5	6	7	8	9	Avg. Diff.
1.0	0.0000	0100	0198	0296	0392	0488	0583	0677	0770	0862	95
1.1	0953	1044	1133	1222	1310	1398	1484	1570	1655	1740	87
1.2	1823	1906	1989	2070	2151	2231	2311	2390	2469	2546	80
1.3	2624	2700	2776	2852	2927	3001	3075	3148	3221	3293	74
1.4	3365	3436	3507	3577	3646	3716	3784	3853	3920	3988	69
1.5	0.4055	4121	4187	4253	4318	4383	4447	4511	4574	4637	65
1.6	4700	4762	4824	4886	4947	5008	5068	5128	5188	5247	61
1.7	5306	5365	5423	5481	5539	5596	5653	5710	5766	5822	57
1.8	5878	5933	5988	6043	6098	6152	6206	6259	6313	6366	54
1.9	6419	6471	6523	6575	6627	6678	6729	6780	6831	6881	51
2.0	0.6931	6981	7031	7080	7129	7178	7227	7275	7324	7372	49
2.1	7419	7467	7514	7561	7608	7655	7701	7747	7793	7839	47
2.2	7885	7930	7975	8020	8065	8109	8154	8198	8242	8286	44
2.3	8329	8372	8416	8459	8502	8544	8587	8629	8671	8713	43
2.4	8755	8796	8838	8879	8920	8961	9002	9042	9083	9123	41
2.5	0.9163	9203	9243	9282	9322	9361	9400	9439	9478	9517	39
2.6	9555	9594	9632	9670	9708	9746	9783	9821	9858	9895	38
2.7	9933	9969	1.0006	1.0043	1.0080	1.0116	1.0152	1.0188	1.0225	1.0260	36
2.8	1.0296	0332	0367	0403	0438	0473	0508	0543	0578	0613	35
2.9	0647	0682	0716	0750	0784	0818	0852	0886	0919	0953	34
3.0	1.0986	1019	1053	1086	1119	1151	1184	1217	1249	1282	33
3.1	1314	1346	1378	1410	1442	1474	1506	1537	1569	1600	32
3.2	1632	1663	1694	1725	1756	1787	1817	1848	1878	1909	31
3.3	1939	1969	2000	2030	2060	2090	2119	2149	2179	2208	30
3.4	2238	2267	2296	2326	2355	2384	2413	2442	2470	2499	29
3.5	1.2528	2556	2585	2613	2641	2669	2698	2726	2754	2782	28
3.6	2809	2837	2865	2892	2920	2947	2975	3002	3029	3056	27
3.7	3083	3110	3137	3164	3191	3218	3244	3271	3297	3324	27
3.8	3350	3376	3403	3429	3455	3481	3507	3533	3558	3584	26
3.9	3610	3635	3661	3686	3712	3737	3762	3788	3813	3838	25
4.0	1.3863	3888	3913	3938	3962	3987	4012	4036	4061	4085	25
4.1	4110	4134	4159	4183	4207	4231	4255	4279	4303	4327	24
4.2	4351	4375	4398	4422	4446	4469	4493	4516	4540	4563	23
4.3	4586	4609	4633	4656	4679	4702	4725	4748	4770	4793	23
4.4	4816	4839	4861	4884	4907	4929	4951	4974	4996	5019	22
4.5	1.5041	5063	5085	5107	5129	5151	5173	5195	5217	5239	22
4.6	5261	5282	5304	5326	5347	5369	5390	5412	5433	5454	21
4.7	5476	5497	5518	5539	5560	5581	5602	5623	5644	5665	21
4.8	5686	5707	5728	5748	5769	5790	5810	5831	5851	5872	20
4.9	5892	5913	5933	5953	5974	5994	6014	6034	6054	6074	20
5.0	1.6094	6114	6134	6154	6174	6194	6214	6233	6253	6273	20
5.1	6292	6312	6332	6351	6371	6390	6409	6429	6448	6467	19
5.2	6487	6506	6525	6544	6563	6582	6601	6620	6639	6658	19
5.3	6677	6696	6715	6734	6752	6771	6790	6808	6827	6845	18
5.4	6864	6882	6901	6919	6938	6956	6974	6993	7011	7029	18
5.5	1.7047	7066	7084	7102	7120	7138	7156	7174	7192	7210	18
5.6	7228	7246	7263	7281	7299	7317	7334	7352	7370	7387	18
5.7	7405	7422	7440	7457	7475	7492	7509	7527	7544	7561	17
5.8	7579	7596	7613	7630	7647	7664	7681	7699	7716	7733	17
5.9	7750	7766	7783	7800	7817	7834	7851	7867	7884	7901	17

TABLE B. (*Concluded*).—NAPIERIAN LOGARITHMS (Log_e).

No.	0	1	2	3	4	5	6	7	8	9	Avg. Diff.
6.0	1.7918	7934	7951	7967	7984	8001	8017	8034	8050	8066	16
6.1	8083	8099	8116	8132	8148	8165	8181	8197	8213	8229	16
6.2	8245	8262	8278	8294	8310	8326	8342	8358	8374	8390	16
6.3	8405	8421	8437	8453	8469	8485	8500	8516	8532	8547	16
6.4	8563	8579	8594	8610	8625	8641	8656	8672	8687	8703	15
6.5	1.8718	8733	8749	8764	8779	8795	8810	8825	8840	8856	15
6.6	8871	8886	8901	8916	8931	8946	8961	8976	8991	9006	15
6.7	9021	9036	9051	9066	9081	9095	9110	9125	9140	9155	15
6.8	9169	9184	9199	9213	9228	9242	9257	9272	9286	9301	15
6.9	9315	9330	9344	9359	9373	9387	9402	9416	9430	9445	14
7.0	1.9459	9473	9488	9502	9516	9530	9544	9559	9573	9587	14
7.1	9601	9615	9629	9643	9657	9671	9685	9699	9713	9727	14
7.2	9741	9755	9769	9782	9796	9810	9824	9838	9851	9865	14
7.3	9879	9892	9906	9920	9933	9947	9961	9974	9988	2.0001	13
7.4	2.0015	0028	0042	0055	0069	0082	0096	0109	0122	0136	13
7.5	2.0149	0162	0176	0189	0202	0215	0229	0242	0255	0268	13
7.6	0281	0295	0308	0321	0334	0347	0360	0373	0386	0399	13
7.7	0412	0425	0438	0451	0464	0477	0490	0503	0516	0528	13
7.8	0541	0554	0567	0580	0592	0605	0618	0631	0643	0656	13
7.9	0669	0681	0694	0707	0719	0732	0744	0757	0769	0782	12
8.0	2.0794	0807	0819	0832	0844	0857	0869	0882	0894	0906	12
8.1	0919	0931	0943	0956	0968	0980	0992	1005	1017	1029	12
8.2	1041	1054	1066	1078	1090	1102	1114	1126	1138	1150	12
8.3	1163	1175	1187	1199	1211	1223	1235	1247	1258	1270	12
8.4	1232	1294	1306	1318	1330	1342	1353	1365	1377	1389	12
8.5	2.1401	1412	1424	1436	1448	1459	1471	1483	1494	1506	12
8.6	1518	1529	1541	1552	1564	1576	1587	1599	1610	1622	12
8.7	1633	1645	1656	1668	1679	1691	1702	1713	1725	1736	11
8.8	1748	1759	1770	1782	1793	1804	1815	1827	1838	1849	11
8.9	1861	1872	1883	1894	1905	1917	1928	1939	1950	1961	11
9.0	2.1972	1983	1994	2006	2017	2028	2039	2050	2061	2072	11
9.1	2083	2094	2105	2116	2127	2138	2148	2159	2170	2181	11
9.2	2192	2203	2214	2225	2235	2246	2257	2268	2279	2289	11
9.3	2300	2311	2322	2332	2343	2354	2364	2375	2386	2396	11
9.4	2407	2418	2428	2439	2450	2460	2471	2481	2492	2502	11
9.5	2.2513	2523	2534	2544	2555	2565	2576	2586	2597	2607	10
9.6	2618	2628	2638	2649	2659	2670	2680	2690	2701	2711	10
9.7	2721	2732	2742	2752	2762	2773	2783	2793	2803	2814	10
9.8	2824	2834	2844	2854	2865	2875	2885	2895	2905	2915	10
9.9	2925	2935	2946	2956	2966	2976	2986	2996	3006	3016	10
10.0	2.3026	3125	3224	3322	3410	3513	3609	3702	3795	3887	9.5

$$\log_{10} a = 0.4343 \log_e a; \quad \log_e a = 2.3026 \log_{10} a; \quad \log_e 0.1 = 0.6974 - 3.$$

Examples: $\log_e 2150 = \log_e 2.15 + 3 \times \log_e 10 = 0.7655 + 3 \times 2.3026$
 $\log_e 0.00215 = \log_e 2.15 + 3 \times \log_e 0.1 = 0.7655 + 3(0.6974 - 3)$

TABLE C.—PROPERTIES OF SATURATED STEAM

Abridged from Keenan's Steam Tables, 1930, by permission of the publishers, The American Society of Mechanical Engineers.

For the saturation temperature and the heat content of the saturated liquid the values have been shortened to the nearest tenth of a unit.

Abs. Press.	Temp.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Lb. per Sq. In.	Deg. Fahr.	Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vaporization	Sat. Vapor
Saturated Liquid and Vapor									
p	$t_{\text{sat.}}$	$\bar{V}_l$	$\bar{V}_{\text{sat.}}$	h_l	$h_{\text{sat.}}$	r	ϕ_l	ϕ_v	$\phi_{\text{sat.}}$
0.0887	32	0.01602	330.1	0.0	1073.4	1073.4	0.0000	2.1834	2.1834
0.1217	40	0.01602	244.5	8.1	1077.1	1069.1	0.0162	2.1397	2.1559
0.1780	50	0.01602	1704.8	18.1	1081.7	1063.6	0.0361	2.0870	2.1231
0.2456 ($\frac{1}{2}$ " Hg)	58.8	0.01603	1256.9	26.9	1085.7	1058.8	0.0533	2.0422	2.0955
0.2561	60	0.01603	1208.0	28.0	1086.2	1058.2	0.0555	2.0364	2.0919
0.3628	70	0.01605	869.0	38.0	1090.8	1052.7	0.0746	1.9877	2.0622
0.4912 (1" Hg)	79.1	0.01607	652.7	47.1	1094.9	1047.8	0.0914	1.9451	2.0365
0.5067	80	0.01607	633.8	48.0	1095.3	1047.3	0.0932	1.9407	2.0340
0.6980	90	0.01610	468.5	58.0	1099.8	1041.8	0.1115	1.8955	2.0070
0.7368 (1 $\frac{1}{2}$ " Hg)	91.7	0.01610	445.3	59.7	1100.6	1040.8	0.1147	1.8877	2.0024
0.9487	100	0.01613	350.8	67.9	1104.2	1036.3	0.1295	1.8518	1.9813
0.9824 (2" Hg)	101.2	0.01613	339.5	69.1	1104.8	1035.7	0.1316	1.8468	1.9784
1	101.8	0.01614	333.9	69.7	1105.0	1035.3	0.1326	1.8442	1.9769
1.274	110	0.01616	265.8	77.9	1108.6	1030.8	0.1472	1.8095	1.9567
1.470	115	0.01618	232.3	82.9	1110.8	1027.9	0.1559	1.7889	1.9448
1.692	120	0.01620	203.6	87.9	1113.0	1025.1	0.1646	1.7685	1.9331
2	126.1	0.01623	173.96	94.0	1115.6	1021.6	0.1750	1.7442	1.9192
3	141.5	0.01630	118.86	109.3	1122.0	1012.7	0.2009	1.6847	1.8856
4	153.0	0.01636	90.74	120.8	1126.8	1005.9	0.2198	1.6420	1.8618
5	162.2	0.01641	73.61	130.1	1130.6	1000.4	0.2348	1.6088	1.8435
6	170.1	0.01645	62.05	137.9	1133.7	995.8	0.2473	1.5814	1.8287
7	176.8	0.01649	53.70	144.7	1136.4	991.7	0.2580	1.5582	1.8162
8	182.9	0.01652	47.39	150.8	1138.9	988.1	0.2674	1.5379	1.8053
9	188.3	0.01656	42.44	156.2	1141.0	984.8	0.2758	1.5200	1.7958
10	193.2	0.01658	38.45	161.1	1143.0	981.8	0.2834	1.5040	1.7874
11	197.7	0.01661	35.17	165.7	1144.8	979.1	0.2903	1.4894	1.7797
12	202.0	0.01664	32.42	169.9	1146.4	976.5	0.2968	1.4760	1.7727
13	205.9	0.01666	30.08	173.8	1147.9	974.1	0.3027	1.4636	1.7663
14	209.6	0.01669	28.06	177.5	1149.3	971.8	0.3082	1.4521	1.7604
14.696	212.0	0.01670	26.82	180.0	1150.2	970.2	0.3119	1.4446	1.7564

TABLE C.—PROPERTIES OF SATURATED STEAM.—(Continued)

Abs. Press.	Temp.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporiza- tion, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Lb. per Sq. In.	Deg. Fahr.	Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vapori- zation	Sat. Vapor
p	$t_{\text{sat.}}$	$\bar{V}_l$	$\bar{V}_{\text{sat.}}$	h_l	$h_{\text{sat.}}$	r	ϕ_l	ϕ_v	$\phi_{\text{sat.}}$
15	213.0	0.01671	26.31	181.0	1150.6	969.6	0.3134	1.4414	1.7548
16	216.3	0.01673	24.76	184.4	1151.8	967.4	0.3184	1.4312	1.7496
17	219.4	0.01676	23.40	187.5	1152.9	965.4	0.3230	1.4218	1.7448
18	222.4	0.01678	22.18	190.5	1154.0	963.5	0.3274	1.4127	1.7402
19	225.2	0.01680	21.08	193.3	1155.0	961.7	0.3316	1.4042	1.7358
20	228.0	0.01682	20.095	196.1	1156.0	959.9	0.3356	1.3960	1.7317
21	230.6	0.01684	19.197	198.7	1156.9	958.2	0.3395	1.3883	1.7278
22	233.1	0.01685	18.380	201.2	1157.8	956.6	0.3431	1.3809	1.7240
23	235.5	0.01687	17.630	203.7	1158.6	955.0	0.3466	1.3738	1.7204
24	237.8	0.01689	16.941	206.1	1159.5	953.4	0.3500	1.3670	1.7170
25	240.1	0.01690	16.306	208.3	1160.2	951.9	0.3533	1.3604	1.7137
26	242.2	0.01692	15.718	210.5	1161.0	950.4	0.3564	1.3542	1.7106
27	244.4	0.01694	15.172	212.7	1161.7	949.0	0.3594	1.3481	1.7075
28	246.4	0.01695	14.664	214.7	1162.4	947.7	0.3624	1.3422	1.7046
29	248.4	0.01697	14.190	216.8	1163.1	946.3	0.3652	1.3365	1.7018
30	250.3	0.01698	13.745	218.7	1163.7	945.0	0.3680	1.3310	1.6990
32	254.0	0.01701	12.940	222.5	1165.0	942.5	0.3732	1.3206	1.6938
34	257.6	0.01704	12.226	226.1	1166.1	940.0	0.3783	1.3107	1.6890
36	260.9	0.01707	11.587	229.5	1167.2	937.7	0.3830	1.3014	1.6844
38	264.2	0.01710	11.015	232.8	1168.3	935.5	0.3876	1.2925	1.6800
40	267.2	0.01712	10.497	235.9	1169.2	933.3	0.3919	1.2840	1.6759
42	270.2	0.01715	10.027	239.0	1170.2	931.2	0.3961	1.2759	1.6720
44	273.1	0.01717	9.599	241.9	1171.1	929.2	0.4000	1.2682	1.6683
46	275.8	0.01719	9.207	244.7	1171.9	927.2	0.4039	1.2608	1.6647
48	278.4	0.01722	8.846	247.4	1172.7	925.4	0.4076	1.2537	1.6613
50	281.0	0.01724	8.514	250.0	1173.5	923.5	0.4111	1.2469	1.6580
52	283.5	0.01726	8.206	252.5	1174.3	921.7	0.4145	1.2404	1.6549
54	285.9	0.01728	7.919	255.0	1175.0	920.0	0.4178	1.2340	1.6518
56	288.2	0.01730	7.653	257.4	1175.7	918.3	0.4210	1.2279	1.6489
58	290.5	0.01732	7.405	259.7	1176.4	916.6	0.4241	1.2220	1.6461
60	292.7	0.01735	7.172	262.0	1177.0	915.0	0.4271	1.2162	1.6434
62	294.8	0.01737	6.955	264.2	1177.6	913.4	0.4300	1.2107	1.6407
64	296.9	0.01739	6.749	266.3	1178.2	911.9	0.4329	1.2053	1.6382
66	299.0	0.01741	6.556	268.4	1178.8	910.4	0.4356	1.2001	1.6357
68	301.0	0.01743	6.375	270.5	1179.4	908.9	0.4384	1.1950	1.6333
70	302.9	0.01744	6.203	272.5	1179.9	907.4	0.4410	1.1900	1.6310
72	304.8	0.01746	6.041	274.5	1180.5	906.0	0.4435	1.1852	1.6287
74	306.7	0.01748	5.887	276.4	1181.0	904.6	0.4460	1.1805	1.6265
76	308.5	0.01750	5.741	278.3	1181.5	903.2	0.4485	1.1759	1.6244
78	310.3	0.01752	5.602	280.1	1182.0	901.9	0.4509	1.1714	1.6223

TABLE C.—PROPERTIES OF SATURATED STEAM.—(Continued)

Abs. Press.	Temp.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Lb. per Sq. In.	Deg. Fahr.	Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vapori- zation	Sat. Vapor
p	$t_{\text{sat.}}$	$\bar{V}_l$	$\bar{V}_{\text{sat.}}$	h_l	$h_{\text{sat.}}$	r	ϕ_l	ϕ_v	$\phi_{\text{sat.}}$
80	312.0	0.01754	5.470	281.9	1182.4	900.5	0.4532	1.1670	1.6202
82	313.7	0.01756	5.343	283.7	1182.9	899.2	0.4555	1.1627	1.6182
84	315.4	0.01757	5.222	285.4	1183.4	897.9	0.4578	1.1586	1.6163
86	317.1	0.01759	5.107	287.1	1183.8	896.7	0.4599	1.1545	1.6144
88	318.7	0.01761	4.997	288.8	1184.2	895.4	0.4621	1.1505	1.6126
90	320.3	0.01763	4.892	290.4	1184.6	894.2	0.4642	1.1465	1.6107
92	321.8	0.01764	4.791	292.1	1185.0	893.0	0.4663	1.1427	1.6090
94	323.4	0.01766	4.694	293.7	1185.4	891.8	0.4683	1.1389	1.6072
96	324.9	0.01768	4.602	295.2	1185.8	890.6	0.4703	1.1352	1.6055
98	326.4	0.01769	4.512	296.8	1186.2	889.4	0.4723	1.1316	1.6038
100	327.8	0.01771	4.426	298.3	1186.6	888.2	0.4742	1.1280	1.6022
102	329.3	0.01773	4.344	299.8	1186.9	887.1	0.4761	1.1245	1.6006
104	330.7	0.01774	4.265	301.3	1187.3	886.0	0.4779	1.1211	1.5990
106	332.1	0.01776	4.189	302.8	1187.6	884.9	0.4798	1.1177	1.5974
108	333.4	0.01777	4.115	304.2	1188.0	883.8	0.4816	1.1144	1.5959
110	334.8	0.01779	4.044	305.6	1188.3	882.7	0.4834	1.1111	1.5944
115	338.1	0.01783	3.878	309.0	1189.1	880.0	0.4876	1.1032	1.5908
120	341.3	0.01786	3.725	312.4	1189.8	877.4	0.4918	1.0956	1.5874
125	344.3	0.01790	3.587	315.6	1190.5	874.9	0.4958	1.0882	1.5840
130	347.3	0.01794	3.451	318.7	1191.2	872.4	0.4996	1.0812	1.5808
135	350.2	0.01797	3.329	321.8	1191.8	870.0	0.5034	1.0743	1.5777
140	353.0	0.01801	3.216	324.7	1192.4	867.7	0.5070	1.0677	1.5747
145	355.8	0.01804	3.110	327.6	1193.0	865.3	0.5105	1.0612	1.5718
150	358.4	0.01808	3.010	330.4	1193.5	863.1	0.5140	1.0550	1.5690
155	361.0	0.01811	2.917	333.2	1194.0	860.8	0.5173	1.0490	1.5663
160	363.5	0.01814	2.830	335.9	1194.5	858.7	0.5205	1.0431	1.5636
165	366.0	0.01818	2.748	338.5	1195.0	856.5	0.5237	1.0374	1.5611
170	368.4	0.01821	2.671	341.0	1195.4	854.4	0.5268	1.0318	1.5586
175	370.8	0.01824	2.598	343.5	1195.9	852.3	0.5298	1.0264	1.5562
180	373.1	0.01827	2.529	346.0	1196.3	850.3	0.5327	1.0211	1.5538
185	375.3	0.01830	2.463	348.4	1196.7	848.2	0.5356	1.0159	1.5515
190	377.5	0.01833	2.401	350.8	1197.0	846.3	0.5384	1.0109	1.5493
195	379.7	0.01836	2.342	353.1	1197.4	844.3	0.5411	1.0060	1.5471
200	381.8	0.01839	2.285	355.3	1197.8	842.4	0.5438	1.0012	1.5450
210	385.9	0.01844	2.180	359.8	1198.4	838.6	0.5491	0.9918	1.5409
220	389.9	0.01850	2.084	364.0	1199.0	835.0	0.5540	0.9829	1.5369
230	393.7	0.01856	1.9964	368.1	1199.6	831.4	0.5588	0.9743	1.5332
240	397.4	0.01861	1.9156	372.1	1200.1	827.9	0.5635	0.9661	1.5295
250	401.0	0.01867	1.8410	376.0	1200.5	824.5	0.5680	0.9581	1.5261
260	404.4	0.01872	1.7723	379.8	1201.0	821.2	0.5723	0.9504	1.5227

TABLE C.—PROPERTIES OF SATURATED STEAM.—(Continued)

Abs. Press.	Temp.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Total Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Lb. per Sq. In.	Deg. Fahr.								
Saturated Liquid and Vapor		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vapori- zation	Sat. Vapor
p	$t_{\text{sat.}}$	$\bar{V}_l$	$\bar{V}_{\text{sat.}}$	h_l	$h_{\text{sat.}}$	r	ϕ_l	ϕ_v	$\phi_{\text{sat.}}$
270	407.8	0.01877	1.7083	383.4	1201.4	818.0	0.5765	0.9430	1.5194
280	411.1	0.01882	1.6490	387.0	1201.8	814.7	0.5805	0.9357	1.5163
290	414.2	0.01887	1.5934	390.5	1202.1	811.6	0.5845	0.9287	1.5132
300	417.3	0.01892	1.5414	393.9	1202.4	808.5	0.5883	0.9220	1.5102
320	423.3	0.01901	1.4469	400.5	1203.0	802.5	0.5957	0.9089	1.5046
340	429.0	0.01910	1.3630	406.8	1203.4	796.6	0.6027	0.8965	1.4992
360	434.4	0.01918	1.2881	412.8	1203.7	790.9	0.6094	0.8846	1.4940
380	439.6	0.01927	1.2208	418.6	1203.9	785.3	0.6157	0.8733	1.4891
400	444.6	0.0194	1.1601	424.2	1204.1	779.8	0.6218	0.8625	1.4843
420	449.4	0.0194	1.1047	429.6	1204.1	774.5	0.6277	0.8520	1.4798
440	454.0	0.0195	1.0540	434.8	1204.1	769.3	0.6334	0.8420	1.4753
460	458.5	0.0196	1.0077	439.9	1204.0	764.1	0.6388	0.8322	1.4711
480	462.8	0.0197	0.9653	444.9	1203.9	759.0	0.6441	0.8228	1.4670
500	467.0	0.0198	0.9261	449.7	1203.7	754.0	0.6493	0.8137	1.4630
600	486.2	0.0202	0.7677	472.3	1202.1	729.8	0.6731	0.7716	1.4447
700	503.0	0.0206	0.6527	492.9	1199.7	706.8	0.6943	0.7342	1.4285
800	518.2	0.0209	0.5653	511.8	1196.7	684.9	0.7135	0.7004	1.4139
900	531.9	0.0213	0.4969	529.5	1193.3	663.8	0.7311	0.6694	1.4005
1000	544.6	0.0217	0.4419	546.0	1189.6	643.5	0.7473	0.6408	1.3881
1100	556.3	0.0222	0.3960	561.7	1185.6	623.9	0.7624	0.6141	1.3765
1200	567.1	0.0226	0.3582	576.5	1181.4	604.9	0.7764	0.5891	1.3656
1300	577.3	0.0230	0.3259	590.6	1177.0	586.3	0.7897	0.5654	1.3552
1400	587.0	0.0235	0.2983	604.3	1172.4	568.1	0.8024	0.5428	1.3452
1500	596.1	0.0239	0.2741	617.5	1167.6	550.2	0.8146	0.5212	1.3357
1600	604.7	0.0244	0.2528	630.2	1162.7	532.6	0.8262	0.5003	1.3265
1800	620.9	0.0254	0.2167	654.7	1151.8	497.2	0.8482	0.4601	1.3083
2000	635.6	0.0265	0.1875	679.0	1139.0	460.0	0.8696	0.4200	1.2896
2200	649.2	0.0277	0.1623	703.7	1123.8	420.0	0.8912	0.3788	1.2700
2400	661.9	0.0292	0.1404	729.4	1105.8	376.4	0.9133	0.3356	1.2488
2600	673.8	0.0310	0.1205	756.7	1084.5	327.8	0.9364	0.2892	1.2257
2800	684.9	0.0333	0.1021	786.7	1058.9	272.3	0.9618	0.2379	1.1996
3000	695.2	0.0367	0.0844	823.1	1025.6	202.5	0.9922	0.1754	1.1676
3200	704.9	0.0459	0.0601	887.0	962.9	75.9	1.0461	0.0651	1.1112
3226*	706.1	0.0522	0.0522	925.0	925.0	0.0	1.0785	0.00	1.0785

* The last line gives the critical values.

TABLE D.—SUPERHEATED STEAM

Abridged from Keenan's Steam Tables, 1930, by permission of the publishers,
The American Society of Mechanical Engineers.

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Sym- bol	Temperature, Degrees Fahrenheit							
		150	200	300	400	500	600	800	1000
1 (101.76)	$\bar{V}$	362.7	392.5	452.1	511.7	571.3	630.9	750.2	869.4
	h	1127.0	1149.8	1195.0	1240.7	1287.2	1334.6	1432.6	1535.2
	ϕ	2.0144	2.0503	2.1142	2.1707	2.2218	2.2688	2.3535	2.4291
2 (126.10)	$\bar{V}$	181.2	196.08	225.9	255.7	285.6	315.4	375.1	434.7
	h	1126.5	1149.4	1194.8	1240.6	1287.1	1334.6	1432.5	1535.2
	ϕ	1.9374	1.9734	2.0376	2.0942	2.1454	2.1924	2.2771	2.3527
5 (162.25)	$\bar{V}$		78.17	90.21	102.19	114.16	126.11	149.99	173.83
	h		1148.2	1194.2	1240.2	1286.9	1334.4	1432.5	1535.1
	ϕ		1.8712	1.9361	1.9930	2.0443	2.0914	2.1761	2.2517
10 (193.21)	$\bar{V}$		38.88	44.98	51.01	57.02	63.01	74.96	86.89
	h		1146.3	1193.2	1239.6	1286.4	1334.0	1432.3	1535.0
	ϕ		1.7925	1.8587	1.9161	1.9676	2.0148	2.0997	2.1753
15 (213.03)	$\bar{V}$			29.90	33.95	37.97	41.98	49.95	57.91
	h			1192.2	1238.9	1286.0	1333.7	1432.1	1534.9
	ϕ			1.8130	1.8708	1.9226	1.9699	2.0550	2.1307
20 (227.96)	$\bar{V}$			22.36	25.43	28.45	31.46	37.44	43.42
	h			1191.1	1238.3	1285.5	1333.4	1432.0	1534.8
	ϕ			1.7802	1.8386	1.8906	1.9380	2.0232	2.0989
25 (240.07)	$\bar{V}$			17.838	20.30	22.73	25.15	29.94	34.73
	h			1190.0	1237.6	1285.1	1333.0	1431.8	1534.7
	ϕ			1.7544	1.8134	1.8657	1.9132	1.9985	2.0743
30 (250.34)	$\bar{V}$			14.818	16.890	18.923	20.94	24.94	28.93
	h			1188.7	1237.0	1284.6	1332.7	1431.6	1534.6
	ϕ			1.7332	1.7928	1.8452	1.8928	1.9783	2.0541
40 (267.24)	$\bar{V}$			11.044	12.623	14.161	15.682	18.686	21.68
	h			1186.5	1235.6	1283.7	1332.0	1431.3	1534.4
	ϕ			1.6990	1.7599	1.8128	1.8607	1.9464	2.0224
50 (281.01)	$\bar{V}$			8.777	10.061	11.304	12.527	14.934	17.337
	h			1183.9	1234.2	1282.8	1331.4	1431.0	1534.1
	ϕ			1.6718	1.7341	1.7875	1.8357	1.9217	1.9977
60 (292.71)	$\bar{V}$			7.260	8.353	9.398	10.423	12.436	14.440
	h			1181.2	1232.8	1281.9	1330.7	1430.6	1533.9
	ϕ			1.6488	1.7128	1.7667	1.8151	1.9014	1.9775
70 (302.92)	$\bar{V}$				7.132	8.036	8.920	10.651	12.370
	h				1231.4	1280.9	1330.0	1430.3	1533.7
	ϕ				1.6945	1.7490	1.7976	1.8843	1.9604
80 (312.03)	$\bar{V}$				6.217	7.015	7.793	9.313	10.817
	h				1229.9	1280.0	1329.3	1429.9	1533.5
	ϕ				1.6785	1.7336	1.7824	1.8694	1.9456

TABLE D.—SUPERHEATED STEAM.—(Continued)

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Sym- bol	Temperature, Degrees Fahrenheit							
		400	450	500	600	700	800	900	1000
100 (327.83)	$\bar{V}$ h ϕ	4.934 1226.9 1.6512	5.263 1252.8 1.6805	5.585 1278.0 1.7075	6.215 1327.9 1.7569	6.831 1378.2 1.8023	7.439 1429.2 1.8445	8.044 1480.7 1.8838	8.644 1533.1 1.9209
120 (341.26)	$\bar{V}$ h ϕ	4.077 1223.8 1.6283	4.359 1250.4 1.6584	4.632 1276.1 1.6859	5.162 1326.5 1.7339	5.680 1377.2 1.7816	6.189 1428.5 1.8240	6.693 1480.2 1.8635	7.196 1532.7 1.9007
140 (353.03)	$\bar{V}$ h ϕ	3.465 1220.5 1.6084	3.713 1247.9 1.6393	3.951 1274.1 1.6674	4.410 1325.1 1.7179	4.857 1376.2 1.7640	5.297 1427.8 1.8066	5.728 1479.7 1.8462	6.161 1532.2 1.8836
160 (363.55)	$\bar{V}$ h ϕ	3.005 1217.1 1.5906	3.227 1245.4 1.6224	3.440 1272.1 1.6510	3.846 1323.7 1.7022	4.240 1375.2 1.7487	4.627 1427.1 1.7915	5.006 1479.1 1.8313	5.385 1531.8 1.8687
180 (373.08)	$\bar{V}$ h ϕ	2.646 1213.5 1.5742	2.849 1242.7 1.6073	3.041 1270.0 1.6364	3.407 1322.2 1.6882	3.760 1374.2 1.7351	4.105 1426.3 1.7782	4.444 1478.5 1.8180	4.782 1531.4 1.8555
200 (381.82)	$\bar{V}$ h ϕ	2.358 1209.8 1.5592	2.547 1240.0 1.5934	2.722 1267.9 1.6231	3.036 1320.8 1.6756	3.376 1373.1 1.7228	3.688 1425.6 1.7662	3.995 1478.0 1.8062	4.299 1531.0 1.8438
220 (389.89)	$\bar{V}$ h ϕ	2.122 1205.9 1.5450	2.299 1237.3 1.5805	2.462 1265.7 1.6109	2.769 1319.3 1.6641	3.062 1372.1 1.7117	3.347 1424.8 1.7553	3.628 1477.4 1.7954	3.903 1530.6 1.8331
250 (400.97)	$\bar{V}$ h ϕ		2.001 1232.9 1.5627	2.149 1262.4 1.5944	2.424 1317.1 1.6483	2.683 1370.5 1.6966	2.938 1423.7 1.7406	3.186 1476.6 1.7810	3.429 1530.0 1.8188
300 (417.33)	$\bar{V}$ h ϕ		1.6347 1225.3 1.5359	1.7648 1256.7 1.5695	2.002 1313.3 1.6256	2.224 1367.8 1.6747	2.438 1421.7 1.7193	2.646 1475.1 1.7601	2.849 1528.9 1.7983
350 (431.71)	$\bar{V}$ h ϕ		1.3712 1217.1 1.5117	1.4899 1250.7 1.5477	1.7003 1309.4 1.6059	1.8945 1365.1 1.6561	2.080 1419.8 1.7012	2.260 1473.6 1.7424	2.435 1527.9 1.7809
400 (444.58)	$\bar{V}$ h ϕ		1.1726 1208.3 1.4892	1.2828 1244.3 1.5276	1.4740 1305.5 1.5884	1.6472 1362.3 1.6396	1.8119 1417.7 1.6854	1.9704 1472.1 1.7270	2.125 1526.8 1.7658
450 (456.27)	$\bar{V}$ h ϕ			1.1204 1237.6 1.5091	1.2972 1301.5 1.5725	1.4548 1359.4 1.6248	1.6032 1415.7 1.6714	1.7455 1470.6 1.7133	1.8834 1525.8 1.7524
500 (466.99)	$\bar{V}$ h ϕ			0.9905 1230.5 1.4915	1.1558 1297.3 1.5579	1.3009 1356.6 1.6113	1.4365 1413.6 1.6586	1.5655 1469.1 1.7009	1.6903 1524.8 1.7404

TABLE D.—SUPERHEATED STEAM.—(Continued)

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Sym- bol	Temperature, Degrees Fahrenheit							
		550	600	650	700	750	800	900	1000
550 (476.92)	$\bar{V}$	0.9658	1.0398	1.1089	1.1746	1.2381	1.2996	1.4181	1.5321
	h	1260.3	1293.1	1323.8	1353.6	1382.8	1411.5	1467.6	1523.8
	ϕ	1.5126	1.5443	1.5726	1.5990	1.6236	1.6468	1.6897	1.7296
600 (486.17)	$\bar{V}$	0.8728	0.9431	1.0080	1.0694	1.1285	1.1855	1.2953	1.4003
	h	1254.6	1288.7	1320.2	1350.6	1380.3	1409.3	1466.1	1522.8
	ϕ	1.4986	1.5316	1.5607	1.5874	1.6125	1.6360	1.6794	1.7196
650 (494.86)	$\bar{V}$	0.7933	0.8609	0.9224	0.9802	1.0355	1.0890	1.1917	1.2886
	h	1248.6	1284.3	1316.6	1347.6	1376.8	1407.1	1464.6	1521.8
	ϕ	1.4851	1.5195	1.5494	1.5766	1.6021	1.6259	1.6698	1.7103
700 (503.04)	$\bar{V}$	0.7251	0.7905	0.8490	0.9038	0.9559	1.0063	1.1029	1.1929
	h	1242.4	1279.7	1312.9	1344.5	1375.1	1404.9	1463.0	1520.8
	ϕ	1.4720	1.5080	1.5387	1.5665	1.5923	1.6165	1.6608	1.7018
750 (510.80)	$\bar{V}$	0.6652	0.7289	0.7852	0.8373	0.8869	0.9348	1.0254	1.1100
	h	1235.8	1275.0	1309.2	1341.4	1372.4	1402.7	1461.5	1519.8
	ϕ	1.4591	1.4970	1.5285	1.5569	1.5831	1.6076	1.6524	1.6938
800 (518.18)	$\bar{V}$	0.6128	0.6750	0.7293	0.7791	0.8265	0.8723	0.9577	1.0374
	h	1228.8	1270.1	1305.4	1338.2	1369.7	1400.4	1459.9	1518.8
	ϕ	1.4462	1.4862	1.5187	1.5477	1.5744	1.5992	1.6446	1.6864
900 (531.95)	$\bar{V}$	0.5234	0.5844	0.6356	0.6821	0.7258	0.7675	0.8451	0.9166
	h	1213.6	1259.8	1297.4	1331.7	1364.2	1395.8	1456.8	1516.8
	ϕ	1.4208	1.4655	1.5003	1.5304	1.5579	1.5835	1.6301	1.6727
1000 (544.58)	$\bar{V}$	0.4495	0.5111	0.5602	0.6040	0.6449	0.6837	0.7547	0.8199
	h	1196.5	1248.7	1289.2	1324.9	1358.5	1391.0	1453.6	1514.8
	ϕ	1.3949	1.4455	1.4829	1.5144	1.5427	1.5691	1.6169	1.6603
1100 (556.28)	$\bar{V}$		0.4500	0.4983	0.5401	0.5788	0.6152	0.6810	0.7408
	h		1236.6	1280.5	1317.9	1352.6	1386.1	1450.4	1512.8
	ϕ		1.4257	1.4662	1.4993	1.5285	1.5557	1.6048	1.6491
1200 (567.14)	$\bar{V}$		0.3985	0.4462	0.4865	0.5233	0.5578	0.6195	0.6750
	h		1223.4	1271.3	1310.6	1346.5	1381.1	1447.2	1510.8
	ϕ		1.4058	1.4501	1.4848	1.5150	1.5431	1.5937	1.6388
1300 (577.32)	$\bar{V}$		0.3537	0.4018	0.4412	0.4767	0.5095	0.5675	0.6190
	h		1208.7	1261.6	1303.0	1340.2	1376.0	1444.0	1508.8
	ϕ		1.3855	1.4343	1.4709	1.5022	1.5312	1.5832	1.6292
1400 (586.96)	$\bar{V}$		0.3144	0.3634	0.4021	0.4365	0.4678	0.5229	0.5712
	h		1192.4	1251.2	1295.2	1333.8	1370.8	1440.8	1506.9
	ϕ		1.3643	1.4187	1.4574	1.4899	1.5199	1.5736	1.6204
1500 (596.08)	$\bar{V}$		0.2789	0.3298	0.3682	0.4014	0.4318	0.4840	0.5298
	h		1174.3	1240.2	1287.1	1327.2	1365.4	1437.6	1504.9
	ϕ		1.3420	1.4029	1.4442	1.4780	1.5091	1.5642	1.6120

TABLE D.—SUPERHEATED STEAM.—(Continued)

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Sym- bol	Temperature, Degrees Fahrenheit							
		650	700	750	800	850	900	950	1000
1600 (604.74)	$\bar{V}$	0.2999	0.3384	0.3708	0.4002	0.4264	0.4503	0.4725	0.4935
	h	1228.3	1278.6	1320.3	1360.0	1398.0	1434.3	1469.3	1502.9
	ϕ	1.3870	1.4313	1.4666	1.4986	1.5283	1.5555	1.5808	1.6042
1700 (612.98)	$\bar{V}$	0.2730	0.3119	0.3437	0.3721	0.3974	0.4204	0.4416	0.4616
	h	1215.5	1269.8	1313.4	1354.4	1393.8	1431.1	1466.8	1500.8
	ϕ	1.3706	1.4185	1.4553	1.4885	1.5250	1.5472	1.5730	1.5968
1800 (620.86)	$\bar{V}$	0.2483	0.2880	0.3195	0.3472	0.3716	0.3938	0.4140	0.4330
	h	1201.5	1260.6	1306.2	1348.7	1389.4	1427.8	1464.3	1498.8
	ϕ	1.3537	1.4058	1.4444	1.4787	1.5104	1.5392	1.5656	1.5897
1900 (628.39)	$\bar{V}$	0.2257	0.2665	0.2977	0.3248	0.3486	0.3700	0.3895	0.4076
	h	1186.0	1250.9	1298.9	1342.9	1385.0	1424.5	1461.7	1496.7
	ϕ	1.3358	1.3931	1.4336	1.4692	1.5019	1.5316	1.5585	1.5830
2000 (635.61)	$\bar{V}$	0.2044	0.2468	0.2781	0.3047	0.3289	0.3486	0.3673	0.3847
	h	1168.9	1240.7	1291.4	1337.0	1380.4	1421.1	1459.2	1494.7
	ϕ	1.3168	1.3802	1.4230	1.4599	1.4957	1.5242	1.5518	1.5765
2200 (649.25)	$\bar{V}$	0.1634	0.2120	0.2438	0.2696	0.2919	0.3115	0.3289	0.3451
	h	1125.8	1218.4	1275.7	1324.8	1371.2	1414.2	1454.0	1490.4
	ϕ	1.2719	1.3536	1.4021	1.4418	1.4779	1.5102	1.5390	1.5644
2400 (661.94)	$\bar{V}$		0.1816	0.2148	0.2401	0.2617	0.2805	0.2969	0.3121
	h		1192.8	1259.1	1312.3	1361.6	1407.1	1448.7	1486.1
	ϕ		1.3253	1.3813	1.4244	1.4628	1.4969	1.5270	1.5531
2600 (673.82)	$\bar{V}$		0.1540	0.1896	0.2151	0.2361	0.2542	0.2699	0.2842
	h		1162.5	1241.0	1299.2	1351.8	1399.8	1443.2	1481.7
	ϕ		1.2938	1.3602	1.4074	1.4483	1.4842	1.5157	1.5426
2800 (684.91)	$\bar{V}$		0.1275	0.1674	0.1934	0.2141	0.2315	0.2466	0.2603
	h		1123.8	1221.3	1285.5	1341.6	1392.1	1437.0	1477.3
	ϕ		1.2559	1.3384	1.3905	1.4342	1.4720	1.5048	1.5326
3000 (695.25)	$\bar{V}$		0.0983	0.1476	0.1742	0.1947	0.2118	0.2265	0.2396
	h		1066.3	1199.3	1271.1	1331.0	1384.3	1431.6	1472.9
	ϕ		1.2028	1.3155	1.3737	1.4203	1.4602	1.4945	1.5233
3200 (704.91)	$\bar{V}$			0.1293	0.1572	0.1777	0.1945	0.2059	0.2214
	h			1174.5	1255.9	1320.1	1376.1	1425.5	1468.4
	ϕ			1.2907	1.3567	1.4067	1.4487	1.4844	1.5144
3226 (706.10) Critical	$\bar{V}$			0.1271	0.1552	0.1757	0.1924	0.2067	0.2192
	h			1171.2	1253.8	1318.7	1375.0	1424.7	1467.8
	ϕ			1.2874	1.3545	1.4050	1.4472	1.4831	1.5133
3500	$\bar{V}$			0.1042	0.1349	0.1556	0.1720	0.1860	0.1981
	h			1129.7	1231.2	1303.0	1363.3	1415.8	1461.6
	ϕ			1.2484	1.3307	1.3866	1.4318	1.4698	1.5017

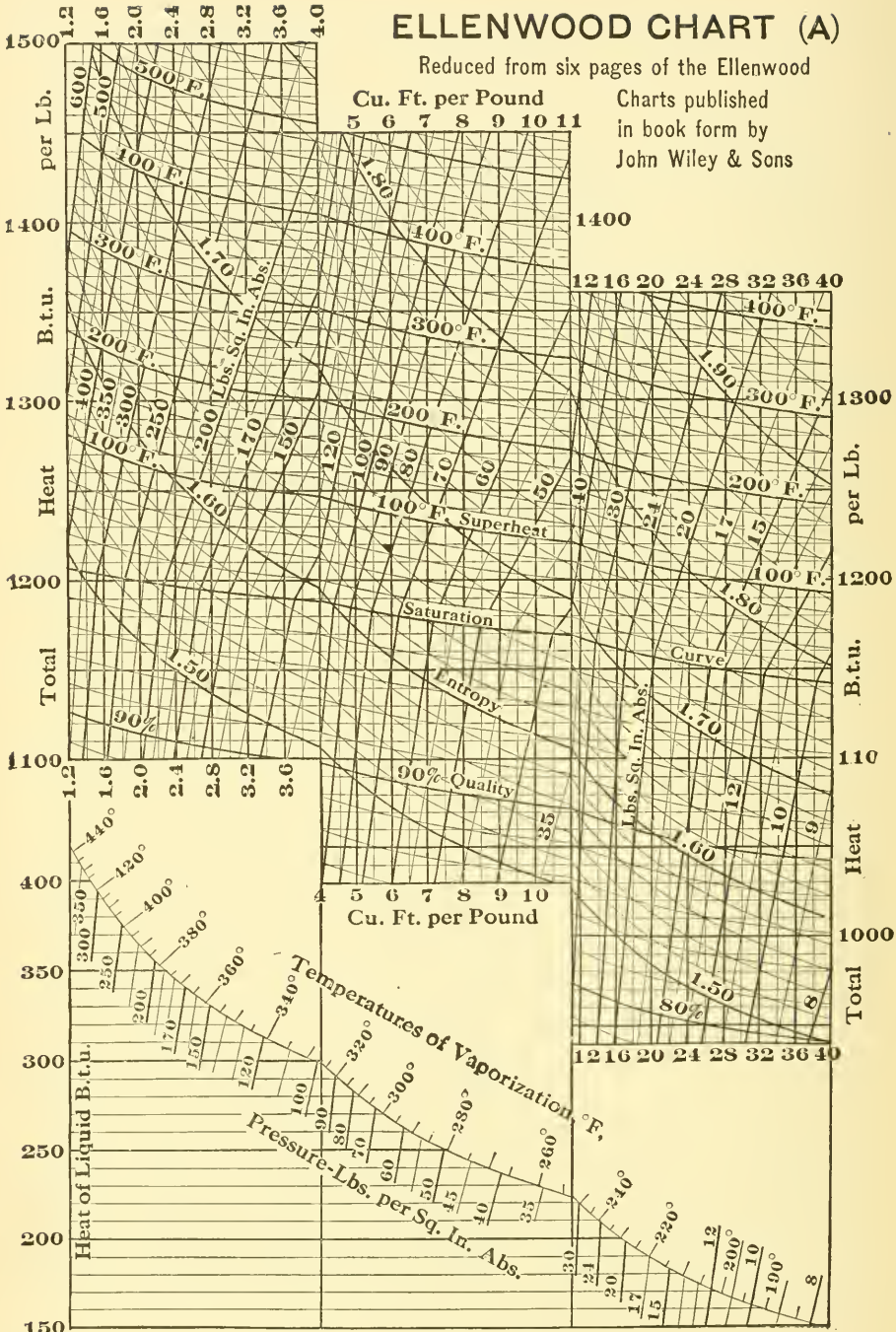


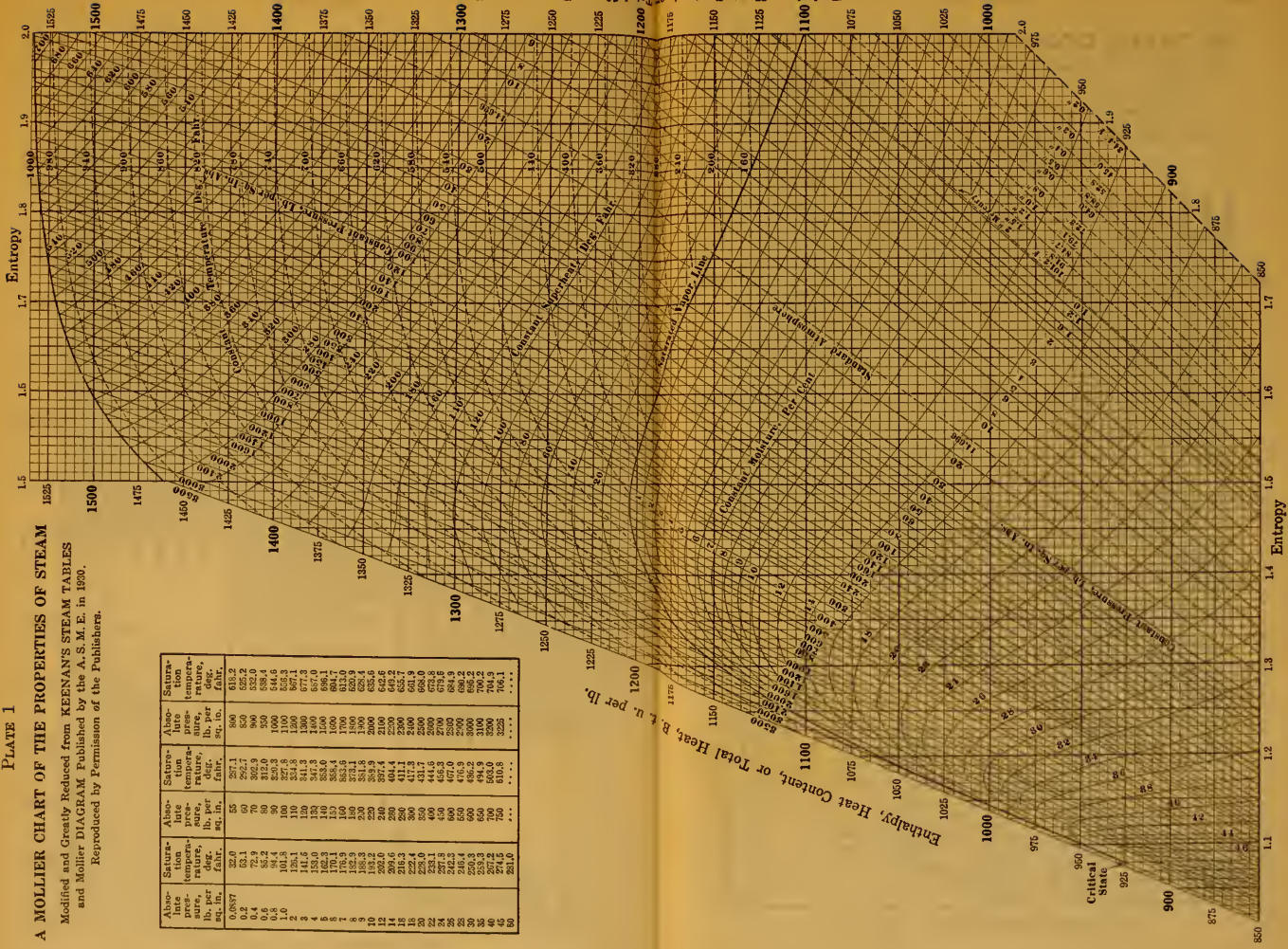
PLATE IIA. THE ELLENWOOD CHART.



A MOLLIER CHART OF THE PROPERTIES OF STEAM

Modified and Greatly Reduced from KEENAN'S STEAM TABLES
and MOLLIER DIAGRAM. Published by the A.S.M.E. in 1930.
Reproduced by Permission of the Publishers.

Absolu- ture, deg. Fahr.	Satura- ture, deg. Fahr.	Absolu- ture, deg. Fahr.	Satura- ture, deg. Fahr.	Absolu- ture, deg. Fahr.	Satura- ture, deg. Fahr.
0.0567	32.0	55	297.1	860	613.2
0.2	53.1	60	292.7	850	595.2
0.4	72.9	70	290.9	800	538.0
0.6	94.4	80	290.3	1000	534.6
0.8	101.5	90	287.8	1100	527.3
1.0	107.3	100	285.3	1200	520.3
2.0	141.5	130	341.3	1300	577.3
3.0	153.0	140	353.3	1400	587.0
4.0	159.4	150	363.3	1500	594.7
5.0	164.1	160	370.3	1600	599.7
6.0	167.4	170	375.3	1700	602.9
7.0	169.8	180	379.3	1800	605.2
8.0	171.4	190	382.3	1900	606.7
9.0	172.3	200	384.3	2000	607.4
10.0	172.8	210	385.3	2100	607.8
11.0	173.0	220	385.3	2200	608.0
12.0	173.0	230	385.3	2300	608.0
13.0	173.0	240	385.3	2400	608.0
14.0	173.0	250	385.3	2500	608.0
15.0	173.0	260	385.3	2600	608.0
16.0	173.0	270	385.3	2700	608.0
17.0	173.0	280	385.3	2800	608.0
18.0	173.0	290	385.3	2900	608.0
19.0	173.0	300	385.3	3000	608.0
20.0	173.0	310	385.3	3100	608.0
21.0	173.0	320	385.3	3200	608.0
22.0	173.0	330	385.3	3300	608.0
23.0	173.0	340	385.3	3400	608.0
24.0	173.0	350	385.3	3500	608.0
25.0	173.0	360	385.3	3600	608.0
26.0	173.0	370	385.3	3700	608.0
27.0	173.0	380	385.3	3800	608.0
28.0	173.0	390	385.3	3900	608.0
29.0	173.0	400	385.3	4000	608.0
30.0	173.0	410	385.3	4100	608.0
31.0	173.0	420	385.3	4200	608.0
32.0	173.0	430	385.3	4300	608.0
33.0	173.0	440	385.3	4400	608.0
34.0	173.0	450	385.3	4500	608.0
35.0	173.0	460	385.3	4600	608.0
36.0	173.0	470	385.3	4700	608.0
37.0	173.0	480	385.3	4800	608.0
38.0	173.0	490	385.3	4900	608.0
39.0	173.0	500	385.3	5000	608.0
40.0	173.0	510	385.3	5100	608.0
41.0	173.0	520	385.3	5200	608.0
42.0	173.0	530	385.3	5300	608.0
43.0	173.0	540	385.3	5400	608.0
44.0	173.0	550	385.3	5500	608.0
45.0	173.0	560	385.3	5600	608.0
46.0	173.0	570	385.3	5700	608.0
47.0	173.0	580	385.3	5800	608.0
48.0	173.0	590	385.3	5900	608.0
49.0	173.0	600	385.3	6000	608.0



ELLENWOOD CHART (B)

Reduced from six pages of the Ellenwood

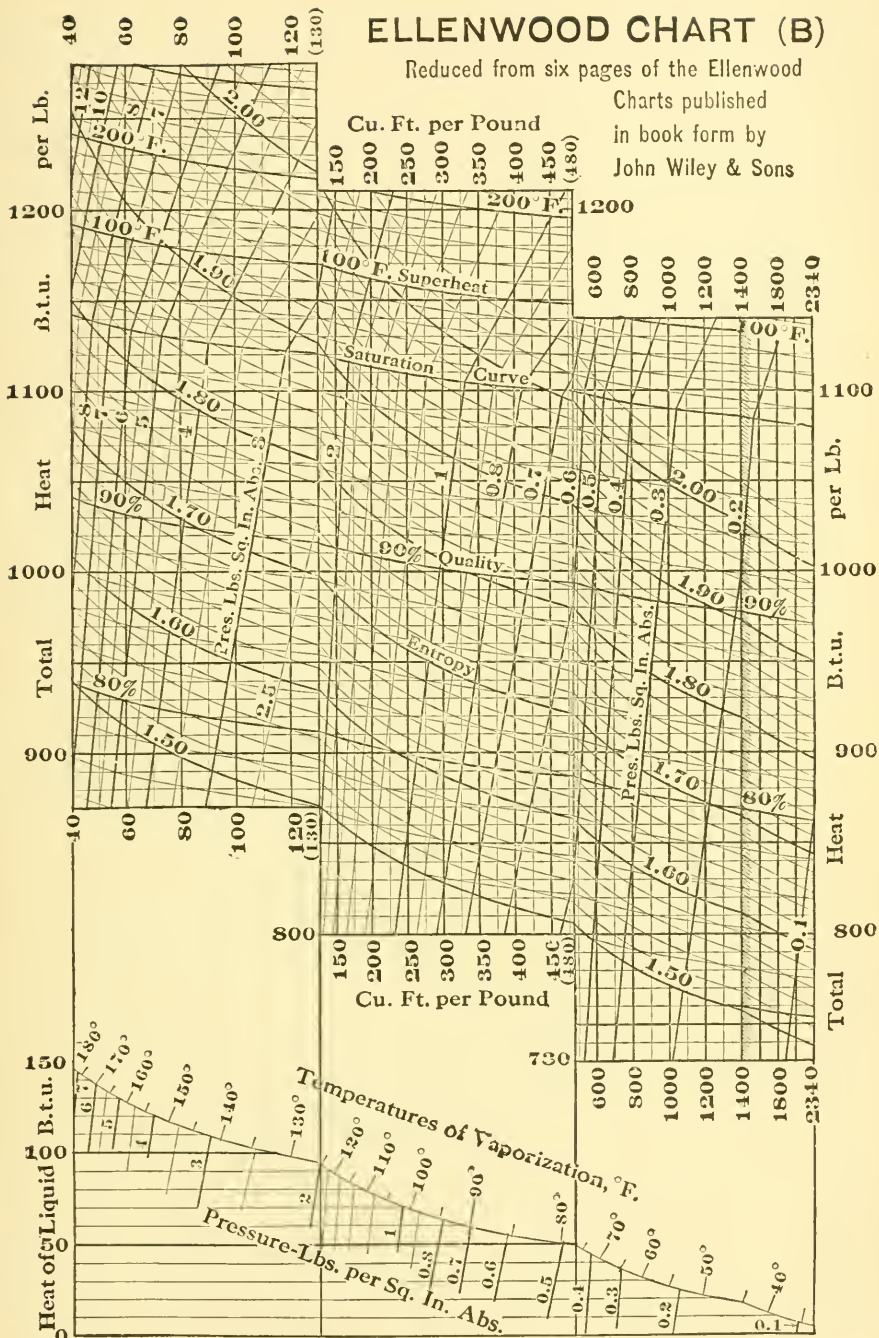
Charts published
in book form by
John Wiley & Sons

PLATE IIB. THE ELLENWOOD CHART (CONT.)

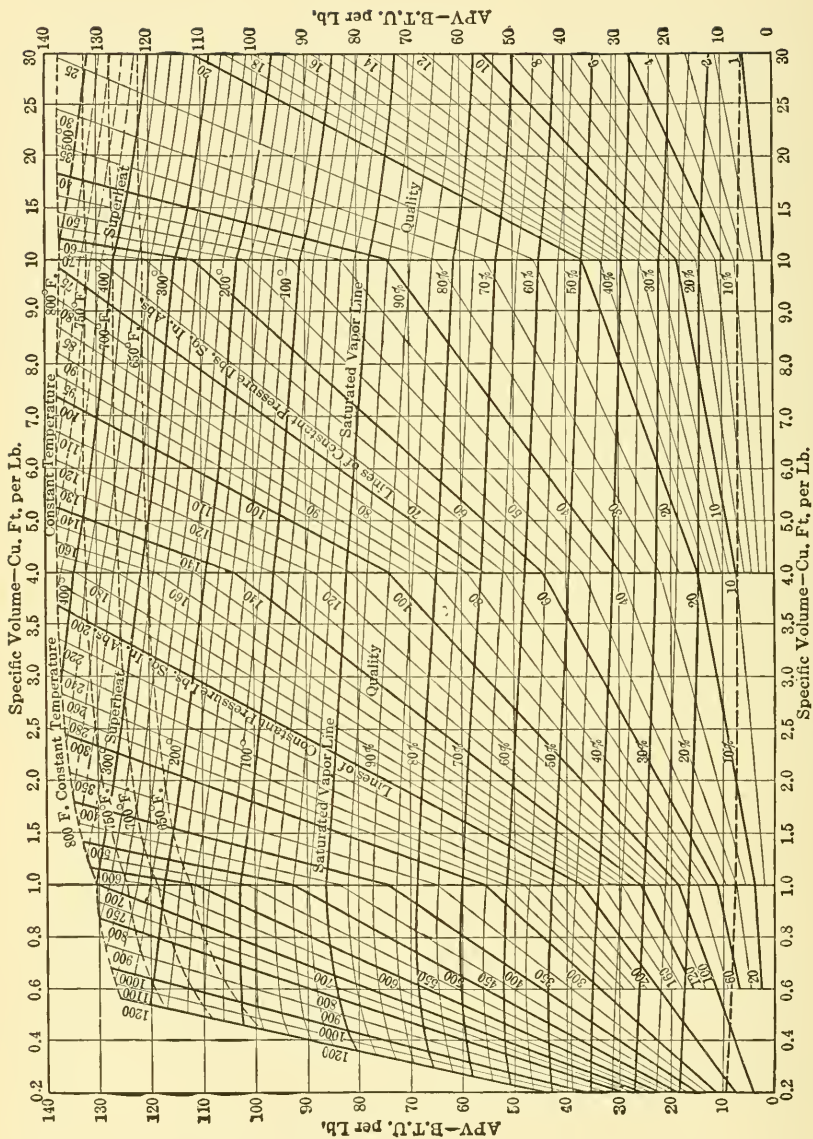


PLATE III. ELLENWOOD APV-V CHART.



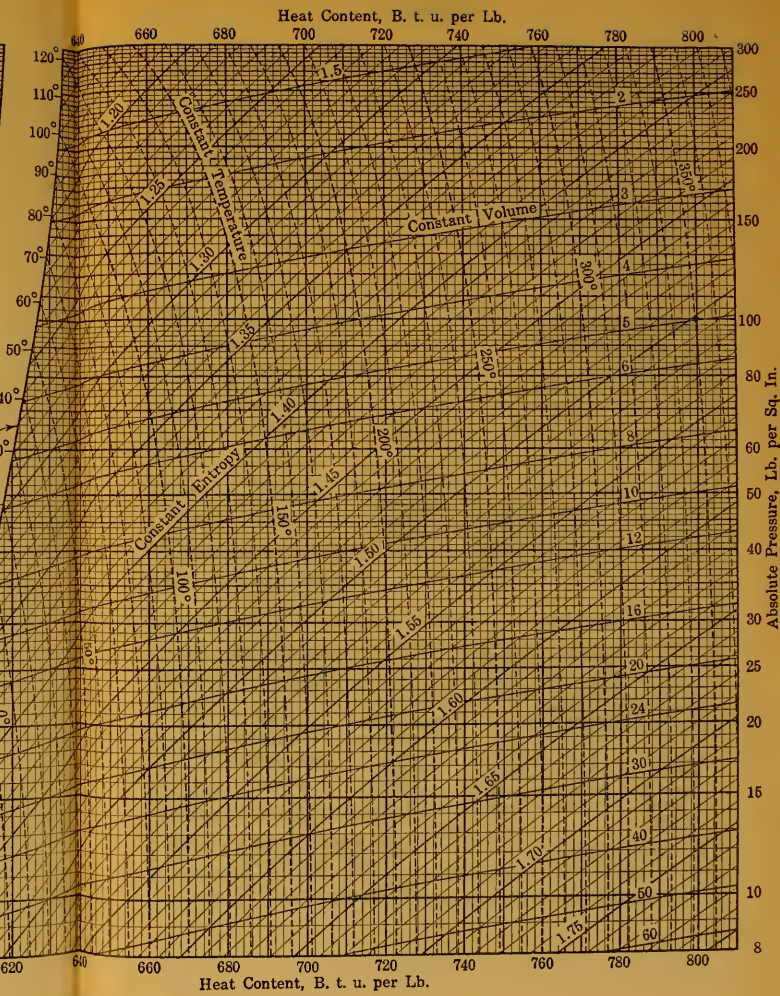
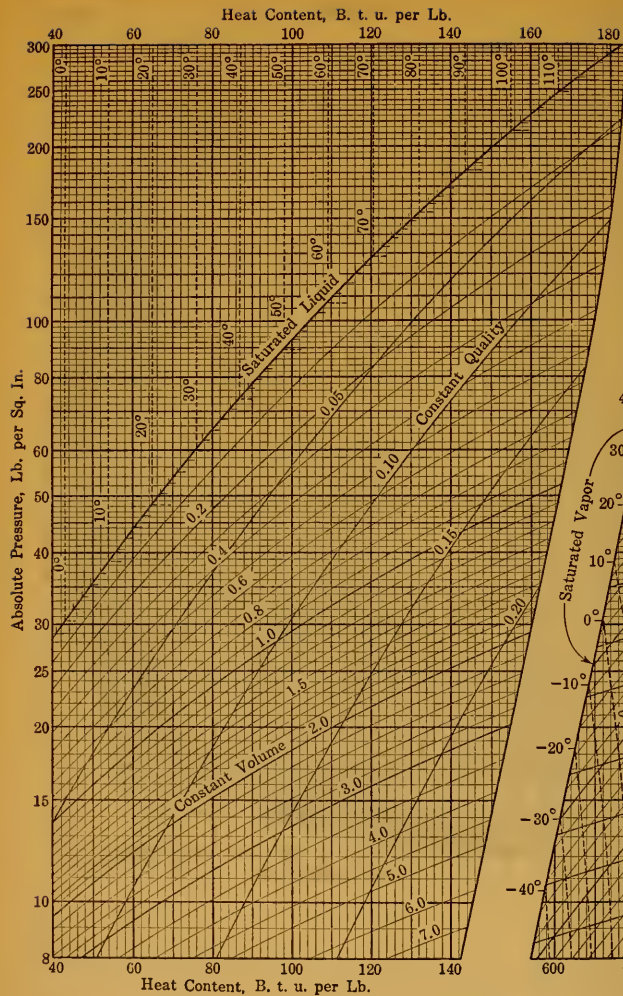


PLATE IV.—A Mollier Chart for Ammonia, Modified from the one in Circular 142, U. S. Bureau of Standards.

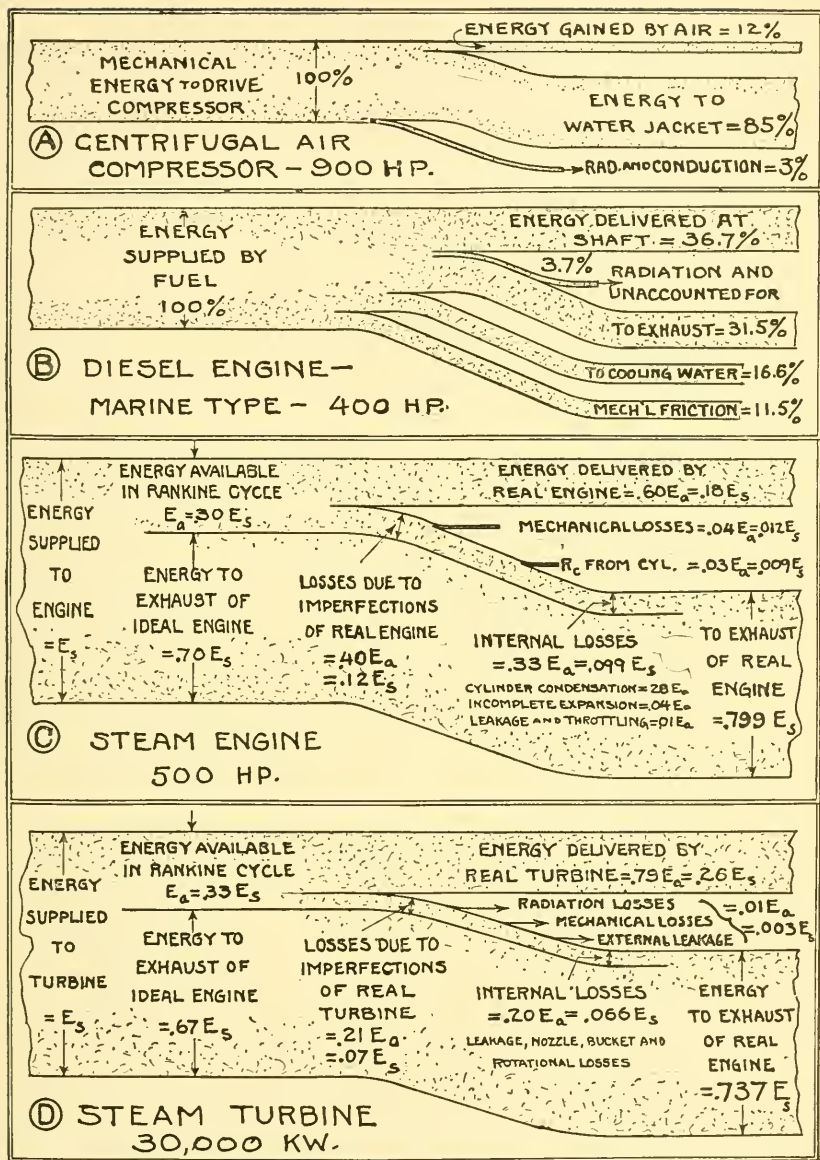


PLATE V. ENERGY STREAMS TYPICAL OF LARGE MODERN MACHINES.

TABLE E.—PROPERTIES OF SATURATED AMMONIA

Abridged from Circular No. 142 U. S. Bureau of Standards

Abs. Press., Lb. per Sq. In.	Temp., Deg. Fahr.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Latent Heat of Vaporization, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
		Saturated Liquid and Vapor	Liquid	Saturated Liquid	Vapor		Saturated Liquid	Vaporization	Saturated Vapor
p	$t_{sat.}$	$\bar{V}_l$	$\bar{V}_{sat.}$	h_l	$h_{sat.}$	r	ϕ_l	ϕ_v	$\phi_{sat.}$
10	-41.34	0.02318	25.81	-1.4	597.1	598.5	-0.0034	1.4310	1.4276
12	-35.16	0.02332	21.77	5.1	599.4	594.3	0.0122	1.4002	1.4124
14	-29.76	0.02345	18.85	10.9	601.4	590.5	0.0257	1.3739	1.3996
16	-24.95	0.02357	16.64	16.0	603.2	587.2	0.0375	1.3510	1.3885
18	-20.61	0.02368	14.90	20.7	604.8	584.1	0.0482	1.3305	1.3787
20	-16.64	0.02377	13.50	25.0	606.2	581.2	0.0578	1.3122	1.3700
22	-12.98	0.02386	12.35	28.9	607.4	578.5	0.0666	1.2955	1.3621
24	-9.58	0.02394	11.39	32.6	608.6	576.0	0.0748	1.2801	1.3549
25	-7.96	0.02398	10.96	34.3	609.1	574.8	0.0787	1.2728	1.3515
26	-6.39	0.02402	10.56	36.0	609.7	573.7	0.0824	1.2658	1.3482
27	-4.87	0.02406	10.20	37.7	610.2	572.5	0.0860	1.2591	1.3451
28	-3.40	0.02410	9.853	39.3	610.7	571.4	0.0895	1.2526	1.3421
29	-1.97	0.02414	9.534	40.8	611.1	570.3	0.0929	1.2463	1.3392
30	-0.57	0.02417	9.236	42.3	611.6	569.3	0.0962	1.2402	1.3364
31	+0.79	0.02421	8.955	43.8	612.0	568.2	0.0993	1.2343	1.3336
32	2.11	0.02425	8.693	45.2	612.4	567.2	0.1024	1.2286	1.3310
33	3.40	0.02428	8.445	46.6	612.8	566.2	0.1055	1.2230	1.3285
34	4.66	0.02432	8.211	48.0	613.2	565.2	0.1084	1.2176	1.3260
34.27	5.00	0.02433	8.150	48.3	613.3	565.0	0.1092	1.2161	1.3253
35	5.89	0.02435	7.991	49.3	613.6	564.3	0.1113	1.2123	1.3236
36	7.09	0.02438	7.782	50.6	614.0	563.4	0.1141	1.2072	1.3213
37	8.27	0.02441	7.584	51.9	614.3	562.4	0.1168	1.2022	1.3190
38	9.42	0.02444	7.396	53.2	614.7	561.5	0.1195	1.1973	1.3168
39	10.55	0.02447	7.217	54.4	615.0	560.6	0.1221	1.1925	1.3146
40	11.66	0.02451	7.047	55.6	615.4	559.8	0.1246	1.1879	1.3125
45	16.88	0.02466	6.307	61.3	616.9	555.6	0.1366	1.1661	1.3027
50	21.67	0.02480	5.710	66.5	618.2	551.7	0.1475	1.1464	1.2939
55	26.09	0.02492	5.218	71.4	619.4	548.0	0.1575	1.1284	1.2859
60	30.21	0.02503	4.805	75.9	620.5	544.6	0.1668	1.1119	1.2787
65	34.06	0.02514	4.453	80.2	621.5	541.3	0.1754	1.0966	1.2720
70	37.70	0.02526	4.151	84.2	622.4	538.2	0.1835	1.0823	1.2658
75	41.13	0.02537	3.887	88.0	623.2	535.2	0.1910	1.0689	1.2599
80	44.40	0.02547	3.655	91.7	624.0	532.3	0.1982	1.0563	1.2545
85	47.50	0.02557	3.449	95.1	624.7	529.6	0.2051	1.0443	1.2494
90	50.47	0.02566	3.266	98.4	625.3	526.9	0.2115	1.0330	1.2445
95	53.32	0.02575	3.101	101.6	625.9	524.3	0.2177	1.0222	1.2399
100	56.05	0.02585	2.952	104.7	626.5	521.8	0.2237	1.0119	1.2356
102	57.11	0.02589	2.896	105.9	626.7	520.8	0.2260	1.0079	1.2339
104	58.16	0.02592	2.843	107.1	626.9	519.8	0.2282	1.0041	1.2323
106	59.19	0.02595	2.791	108.3	627.1	518.8	0.2305	1.0002	1.2307

TABLE E.—PROPERTIES OF SATURATED AMMONIA.—(Continued)

Abs. Press., Lb. per Sq. In.	Temp., Deg. Fahr.	Volume, Cu. Ft. per Lb.		Heat Content, B.t.u. per Lb.		Latent Heat of Vaporiz- ation, B.t.u. per Lb.	Entropy, B.t.u. per Deg. Fahr. per Lb.		
Saturated Liquid and Vapor		Sat. Liquid	Sat. Vapor	Sat. Liquid	Sat. Vapor		Sat. Liquid	Vapori- zation	Sat. Vapor
p	$t_{sat.}$	$\bar{V}_l$	$\bar{V}_{sat.}$	h_l	$h_{sat.}$	r	ϕ_l	ϕ_v	$\phi_{sat.}$
108	60.21	0.02599	2.741	109.4	627.3	517.9	0.2327	0.9964	1.2291
110	61.21	0.02602	2.693	110.5	627.5	517.0	0.2348	0.9927	1.2275
112	62.20	0.02605	2.647	111.7	627.7	516.0	0.2369	0.9890	1.2259
114	63.17	0.02609	2.602	112.8	627.9	515.1	0.2390	0.9854	1.2244
116	64.13	0.02612	2.559	113.9	628.1	514.2	0.2411	0.9819	1.2230
118	65.08	0.02615	2.517	114.9	628.2	513.3	0.2431	0.9784	1.2215
120	66.02	0.02618	2.476	116.0	628.4	512.4	0.2452	0.9749	1.2201
122	66.94	0.02622	2.437	117.1	628.6	511.5	0.2471	0.9715	1.2186
124	67.86	0.02625	2.399	118.1	628.7	510.6	0.2491	0.9682	1.2173
126	68.76	0.02628	2.362	119.1	628.9	509.8	0.2510	0.9649	1.2159
128	69.65	0.02631	2.326	120.1	629.0	508.9	0.2529	0.9616	1.2145
130	70.53	0.02634	2.291	121.1	629.2	508.1	0.2548	0.9584	1.2132
132	71.40	0.02637	2.258	122.1	629.3	507.2	0.2567	0.9552	1.2119
134	72.26	0.02640	2.225	123.1	629.5	506.4	0.2585	0.9521	1.2106
136	73.11	0.02643	2.193	124.1	629.6	505.5	0.2603	0.9490	1.2093
138	73.95	0.02646	2.162	125.1	629.8	504.7	0.2621	0.9460	1.2081
140	74.79	0.02649	2.132	126.0	629.9	503.9	0.2638	0.9430	1.2068
145	76.82	0.02656	2.061	128.3	630.2	501.9	0.2681	0.9357	1.2038
150	78.81	0.02664	1.994	130.6	630.5	499.9	0.2724	0.9285	1.2009
155	80.75	0.02671	1.931	132.8	630.8	498.0	0.2764	0.9215	1.1979
160	82.64	0.02678	1.872	135.0	631.1	496.1	0.2804	0.9148	1.1952
165	84.48	0.02685	1.816	137.2	631.3	494.1	0.2842	0.9083	1.1925
169.2	86.00	0.02691	1.772	138.9	631.5	492.6	0.2875	0.9029	1.1904
170	86.29	0.02692	1.764	139.3	631.6	492.3	0.2881	0.9019	1.1900
175	88.05	0.02699	1.714	141.3	631.7	490.4	0.2917	0.8957	1.1874
180	89.78	0.02706	1.667	143.3	632.0	488.7	0.2954	0.8896	1.1850
185	91.47	0.02712	1.623	145.2	632.1	486.9	0.2989	0.8837	1.1826
190	93.13	0.02719	1.581	147.2	632.4	485.2	0.3024	0.8778	1.1802
195	94.75	0.02726	1.540	149.1	632.5	483.4	0.3057	0.8722	1.1779
200	96.34	0.02732	1.502	150.9	632.7	481.8	0.3090	0.8666	1.1756
210	99.43	0.02745	1.431	154.6	633.0	478.4	0.3154	0.8559	1.1713
220	102.42	0.02757	1.367	158.0	633.2	475.2	0.3216	0.8455	1.1671
230	105.30	0.02770	1.307	161.4	633.4	472.0	0.3275	0.8356	1.1631
240	108.09	0.02782	1.253	164.7	633.6	468.9	0.3332	0.8260	1.1592
250	110.80	0.02794	1.202	168.0	633.8	465.8	0.3388	0.8167	1.1555
260	113.42	0.02806	1.155	171.1	633.9	462.8	0.3441	0.8077	1.1518
270	115.97	0.02818	1.112	174.1	633.9	459.8	0.3494	0.7989	1.1483
280	118.45	0.02829	1.072	177.1	634.0	456.9	0.3545	0.7904	1.1449
290	120.86	0.02840	1.034	180.0	634.0	454.0	0.3594	0.7821	1.1415
300	123.21	0.02852	0.999	182.9	634.0	451.1	0.3642	0.7741	1.1383

TABLE F.—PROPERTIES OF SUPERHEATED AMMONIA

Abridged from Circular No. 142 U. S. Bureau of Standards

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Sym- bol	Temperature, Degrees Fahrenheit							
		0	20	40	60	80	100	150	200
8 (-48.64)	$\bar{V}$	35.83	37.45	39.07	40.68	42.28	43.88	47.87	
	h	619.5	629.7	639.8	649.9	660.1	670.3	696.1	
	ϕ	1.5044	1.5261	1.5467	1.5666	1.5858	1.6044	1.6485	
10 (-41.34)	$\bar{V}$	28.58	29.90	31.20	32.49	33.78	35.07	38.26	
	h	618.9	629.1	639.3	649.5	659.7	670.0	695.8	
	ϕ	1.4773	1.4992	1.5200	1.5400	1.5593	1.5779	1.6222	
15 (-27.29)	$\bar{V}$	18.92	19.82	20.70	21.58	22.44	23.31	25.46	27.59
	h	617.2	627.8	638.2	648.5	658.9	669.2	695.3	721.7
	ϕ	1.4272	1.4497	1.4709	1.4912	1.5108	1.5296	1.5742	1.6158
20 (-16.64)	$\bar{V}$	14.09	14.78	15.45	16.12	16.78	17.43	19.05	20.66
	h	615.5	626.4	637.0	647.5	658.0	668.5	694.7	721.2
	ϕ	1.3907	1.4138	1.4356	1.4562	1.4760	1.4950	1.5399	1.5817
25 (-7.96)	$\bar{V}$	11.19	11.75	12.30	12.84	13.37	13.90	15.21	16.50
	h	613.8	625.0	635.8	646.5	657.1	667.7	694.1	720.8
	ϕ	1.3616	1.3855	1.4077	1.4287	1.4487	1.4679	1.5131	1.5552
30 (-0.57)	$\bar{V}$	9.250	9.731	10.20	10.65	11.10	11.55	12.65	13.73
	h	611.9	623.5	634.6	645.5	656.2	666.9	693.5	720.3
	ϕ	1.3371	1.3618	1.3845	1.4059	1.4261	1.4456	1.4911	1.5334
35 (5.89)	$\bar{V}$		8.287	8.695	9.093	9.484	9.869	10.82	11.75
	h		622.0	633.4	644.4	655.3	666.1	692.9	719.9
	ϕ		1.3413	1.3646	1.3863	1.4069	1.4265	1.4724	1.5148
40 (11.66)	$\bar{V}$		7.203	7.568	7.922	8.268	8.609	9.444	10.27
	h		620.4	632.1	643.4	654.4	665.3	692.3	719.4
	ϕ		1.3231	1.3470	1.3692	1.3900	1.4098	1.4561	1.4987
50 (21.67)	$\bar{V}$			5.988	6.280	6.564	6.843	7.521	8.185
	h			629.5	641.2	652.6	663.7	691.1	718.5
	ϕ			1.3169	1.3399	1.3613	1.3816	1.4286	1.4716
60 (30.21)	$\bar{V}$			4.933	5.184	5.428	5.665	6.239	6.798
	h			626.8	639.0	650.7	662.1	689.9	717.5
	ϕ			1.2913	1.3152	1.3373	1.3581	1.4058	1.4493
70 (37.70)	$\bar{V}$			4.177	4.401	4.615	4.822	5.323	5.807
	h			623.9	636.6	648.7	660.4	688.7	716.6
	ϕ			1.2688	1.2937	1.3166	1.3378	1.3863	1.4302
80 (44.40)	$\bar{V}$				3.812	4.005	4.190	4.635	5.063
	h				634.3	646.7	658.7	687.5	715.6
	ϕ				1.2745	1.2981	1.3199	1.3692	1.4136
90 (50.47)	$\bar{V}$				3.353	3.529	3.698	4.100	4.484
	h				631.8	644.7	657.0	686.3	714.7
	ϕ				1.2571	1.2814	1.3038	1.3539	1.3988

TABLE F.—PROPERTIES OF SUPERHEATED AMMONIA.—(Continued)

Abs. Press., Lb. per Sq. In. (Sat. Temp.)	Sym- bol.	Temperature, Degrees Fahrenheit							
		100	120	140	160	180	200	250	300
100 (56.05)	$\bar{V}$	3.304	3.454	3.600	3.743	3.883	4.021	4.361	4.695
	h	655.2	667.3	679.2	690.8	702.3	713.7	742.2	770.8
	ϕ	1.2891	1.3104	1.3305	1.3495	1.3678	1.3854	1.4271	1.4660
110 (61.21)	$\bar{V}$	2.981	3.120	3.255	3.386	3.515	3.642	3.954	4.259
	h	653.4	665.8	677.8	689.6	701.2	712.8	741.5	770.2
	ϕ	1.2755	1.2972	1.3176	1.3370	1.3555	1.3732	1.4151	1.4543
120 (66.02)	$\bar{V}$	2.712	2.842	2.967	3.089	3.209	3.326	3.614	3.895
	h	651.6	664.2	676.5	688.4	700.2	711.8	740.7	769.6
	ϕ	1.2628	1.2850	1.3058	1.3254	1.3441	1.3620	1.4042	1.4435
130 (70.53)	$\bar{V}$	2.484	2.606	2.724	2.838	2.949	3.059	3.326	3.587
	h	649.7	662.7	675.1	687.2	699.1	710.9	739.9	769.0
	ϕ	1.2509	1.2736	1.2947	1.3146	1.3335	1.3516	1.3941	1.4336
140 (74.79)	$\bar{V}$	2.288	2.404	2.515	2.622	2.727	2.830	3.080	3.323
	h	647.8	661.1	673.7	686.0	698.0	709.9	739.2	768.3
	ϕ	1.2396	1.2628	1.2843	1.3045	1.3236	1.3418	1.3846	1.4243
150 (78.81)	$\bar{V}$	2.118	2.228	2.334	2.435	2.534	2.631	2.866	3.095
	h	645.9	659.4	672.3	684.8	696.9	708.9	738.4	767.7
	ϕ	1.2289	1.2526	1.2745	1.2949	1.3142	1.3327	1.3758	1.4157
160 (82.64)	$\bar{V}$	1.969	2.075	2.175	2.272	2.365	2.457	2.679	2.895
	h	643.9	657.8	670.9	683.5	695.8	707.9	737.6	767.1
	ϕ	1.2186	1.2429	1.2652	1.2859	1.3054	1.3240	1.3675	1.4076
170 (86.29)	$\bar{V}$	1.837	1.939	2.035	2.127	2.216	2.303	2.514	2.718
	h	641.9	656.1	669.4	682.3	694.7	706.9	736.8	766.4
	ϕ	1.2087	1.2336	1.2563	1.2773	1.2971	1.3159	1.3596	1.3999
180 (89.78)	$\bar{V}$	1.720	1.818	1.910	1.999	2.084	2.167	2.367	2.561
	h	639.9	654.4	668.0	681.0	693.6	705.9	736.1	765.8
	ϕ	1.1992	1.2247	1.2477	1.2691	1.2891	1.3081	1.3521	1.3926
190 (93.13)	$\bar{V}$	1.615	1.710	1.799	1.884	1.966	2.045	2.236	2.421
	h	637.8	652.6	666.5	679.7	692.5	704.9	735.3	765.2
	ϕ	1.1899	1.2160	1.2396	1.2612	1.2815	1.3007	1.3450	1.3857
200 (96.34)	$\bar{V}$	1.520	1.612	1.698	1.780	1.859	1.935	2.118	2.295
	h	635.6	650.9	665.0	678.4	691.3	703.9	734.5	764.5
	ϕ	1.1809	1.2077	1.2317	1.2537	1.2742	1.2935	1.3382	1.3791
250 (110.80)	$\bar{V}$		1.240	1.316	1.386	1.453	1.518	1.670	1.815
	h		641.5	657.2	671.8	685.5	698.8	730.5	761.3
	ϕ		1.1690	1.1956	1.2195	1.2414	1.2617	1.3081	1.3501
300 (123.21)	$\bar{V}$			1.058	1.123	1.183	1.239	1.372	1.496
	h			648.7	664.7	679.5	693.5	726.5	758.1
	ϕ			1.1632	1.1894	1.2129	1.2344	1.2827	1.3257

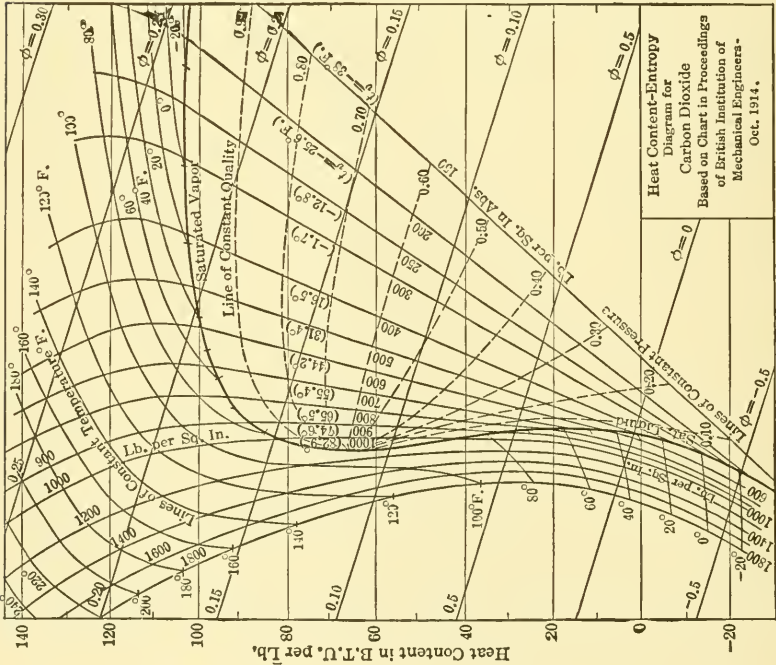


PLATE VI.

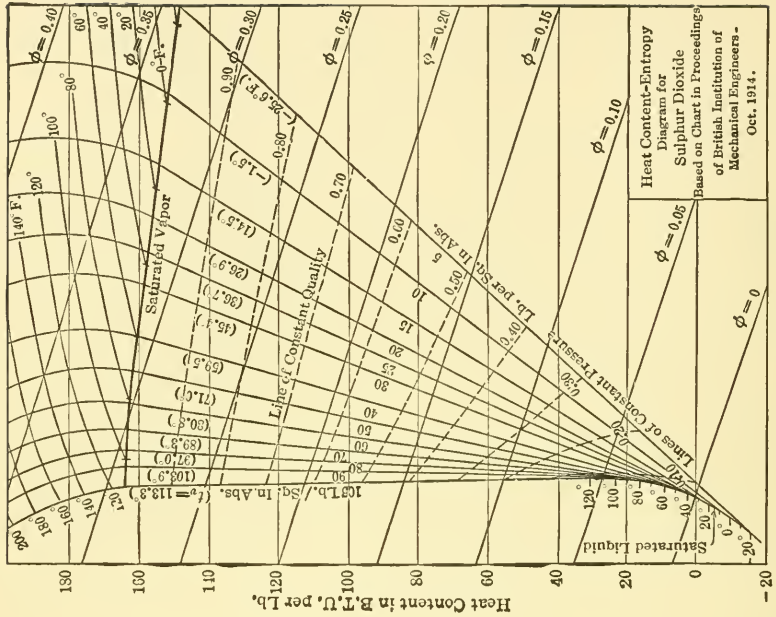


PLATE VII.

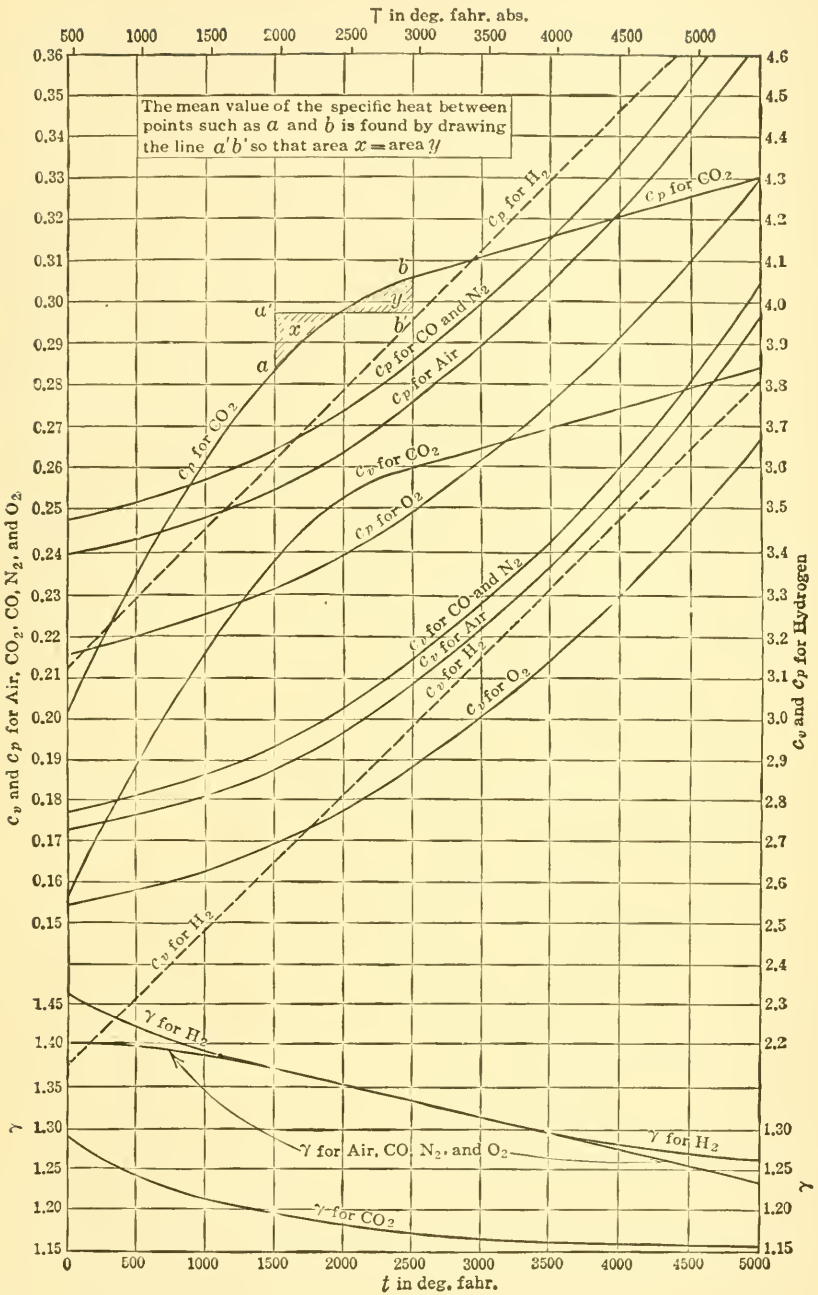


PLATE VIII.—Chart Showing how the Instantaneous Values of c_p , c_v and γ for Certain Gases Vary with the Temperature.

TABLE G.—INSTANTANEOUS SPECIFIC HEATS OF GASES

(In B.t.u. per pound per degree Fahrenheit)

$$C_p = M' + BT + CT^2$$

$$C_v = M + BT + CT^2$$

in which T = abs. temp. in deg. fahr., and the constants have the following values:

Gas	M'	M	B	C	For $T=492$		For $T=2000$	
					c_p	c_v	c_p	c_v
CO	0.2475	0.1765	0	$4.23 (10)^{-3}$	0.2485	0.1775	0.265	0.194
H ₂	2.98	2.00	$3.31 (10)^{-4}$	0	3.140	2.160	3.64	2.66
N ₂	0.2475	0.1765	0	$4.28 (10)^{-3}$	0.2485	0.1775	0.265	0.194
O ₂	0.216	0.154	0	$3.75 (10)^{-3}$	0.217	0.155	0.231	0.169
Air	0.239	0.1705	0	$4.138 (10)^{-3}$	0.240	0.1715	0.2555	0.187
CO ₂ ($T < 2900$)	0.1625	0.1175	$8.86 (10)^{-5}$	$-1.36 (10)^{-8}$	0.2025	0.1575	0.285	0.240
CO ₂ ($T > 2900$)	0.277	0.232	$9.55 (10)^{-6}$	0				
C ₂ H ₂	0.238	0.161	$3.11 (10)^{-4}$	0	0.391	0.314	0.860	0.783
CH ₄	0.216	0.092	$6.59 (10)^{-4}$	0	0.540	0.416	1.53	1.41
C ₂ H ₄	0.238	0.167	$2.42 (10)^{-4}$	0	0.357	0.286	0.722	0.651

The values of M' , M , B and C for air are those calculated in Part I, p. 86. For the other gases the values of these constants are based on the ones of Goodenough and Felbeck, as given on page 106, Bulletin No. 139, University of Illinois Experiment Station.

PROBLEMS

CHAPTER XLII

1. Discuss, in a general way, the engineering problems involved in the flow of fluids through pipes and passages.

2. What are the possible sources of energy to make a fluid flow from one point to another in a closed conduit?

3. (a) Under what conditions would the pressure p_d be greater than p_a for the case indicated in Fig. 829? (b) Derive the energy equation for this case.

4. Discuss the significance of the term x in Eq. (590b).

5. (a) What is Bernoulli's theorem? (b) Its significance? (c) Give some illustrations of flow for which it does not apply.

6. What are the effects of friction and turbulence on a fluid flowing through an insulated pipe line?

7. Discuss the significance of the results shown by the examples given in Sec. 498(d).

8. Why is it so difficult to calculate the friction loss caused by the flow of a gas through a passage where rapid rates of heat transfer are involved?

9. (a) What is meant by the terms "short" and "long" lines, when calculating the drop in pressure due to friction? (b) Derive the expression for this drop in a "short" line.

10. For what conditions does Eq. (591b) hold?

11. What determines the value of the coefficient c in Eq. (591)?

12. Define (a) absolute viscosity, (b) kinematic viscosity, and (c) relative viscosity.

13. If the absolute viscosity of a fluid is 20 centipoises, what is its viscosity (a) in poundal sec. per sq. ft., and (b) in lb. force sec. per sq. ft.

14. Using the Sutherland formula and Fig. 832, calculate the absolute viscosity of CO_2 at atmospheric pressure and a temperature of 1500 deg. fahr.

15. Discuss the effect of temperature and pressure on the absolute viscosity of gases, vapors, and liquids.

16. (a) What is Reynolds' number? (b) Is this an important term? Why?

17. (a) Find Reynolds' number for water at a temperature of 210 deg. fahr. flowing through a 6-in. standard pipe (see p. 825) with a velocity of 240 ft. per min. (b) The same as (a) except that the fluid is superheated steam at a temperature of 500 deg. fahr., an absolute pressure of 100 lb. per sq. in., and a velocity of 12,000 ft. per min. (c) The same as (b) except that the fluid is air.

Ans. (a) $6.52(10)^5$. (b) $13.69(10)^5$. (c) $2.98(10)^5$.

18. (a) What is viscous flow? (b) What is the relation between the average and maximum velocities for such flow?

19. What is Poiseuille's law?

20. (a) Explain what is the general meaning of the term *friction factor*, when dealing with the flow of fluids. (b) Under what conditions is this factor equal to $64 \div \text{Reynolds' number}$? (c) Discuss the relation between Eqs. (591a), (606), and (608).

21. (a) If a fluid is flowing in a pipe line under such conditions that Reynolds'

number is 500, is the flow viscous or turbulent? (b) Do most of the flow problems in power plant work involve viscous or turbulent flow? Why?

22. In Fig. 841, discuss the significance of (a) the main curves, (b) the small right-hand one, and (c) the top one.

23. Find the drop in pressure of a fluid flowing through 100 ft. of insulated horizontal brass tubing, 2 in. inside diameter with $p_1 = 100$ lb. per sq. in. abs., $t_1 = 700$ deg. fahr., and $v = 6000$ ft. per min., if the fluid is (a) air, and (b) steam.

Ans. (a) 2.45 lb. per sq. in. (b) 1.61 lb. per sq. in.

24. Find the rate of flow, in pounds per hour, for each case in problem 23, and check the drop in pressure by a formula that involves the rate of flow.

Ans. (a) 1805. (b) 1138.

25. Under what conditions is the friction factor dependent on the pipe diameter? Discuss.

26. Find the drop in pressure of oil flowing through 1000 ft. of pipe line having an inside diameter of 7.98 in., if the oil has a temperature of 60 deg. fahr., a density of 23.8 deg. A.P.I., an absolute viscosity of 220 centipoises, and a rate of flow of (a) 1000 bbl. per hr.; (b) 2000 bbl. per hr.

Ans. (a) 10.35 lb. per sq. in. (b) 34.8 lb. per sq. in.

27. (a) What is the significance of the curves given in Fig. 842? (b) The origin?

28. (a) Find the drop in pressure of steam flowing through a new insulated horizontal steel pipe having an inside diameter of 6.25 in. and a length of 200 ft. The steam in this line has an average pressure of 1000 lb. per sq. in. abs., an average temperature of 700 deg. fahr., and an average velocity of 12,000 ft. per min. (b) Find the rate of flow in pounds per hour for these conditions.

Ans. (a) 37.5 lb. per sq. in. (b) 254,000.

29. The same as problem 28, except that the steam has a pressure of 1200 lb. per sq. in. abs.

Ans. (a) 46.1 lb. per sq. in. (b) 315,000.

30. What is the chief cause of a pressure drop through a fitting or valve?

31. Find the length of 10-in. straight pipe having a resistance equivalent to that of a 90-deg. bend with a radius of 25 in.

32. (a) Discuss Foster's equation and its application. (b) Discuss the curves given in Fig. 843.

33. A standard screwed elbow will cause a resistance equivalent to about how many pipe diameters for (a) turbulent flow, and (b) viscous flow?

34. (a) Make a sketch of some welded forms of elbows; and (b) compare their resistances to that of the standard screwed one.

35. (a) Discuss the results determined by testing the apparatus shown in Fig. 844. (b) Discuss the effect of square corners in elbows.

36. (a) Give some examples of the use of rough closed conduits. (b) Why is the degree of roughness of a passage difficult to specify? (c) Explain the basis for the curves given in Fig. 847.

37. (a) The same as the problem given at the bottom of p. 817, except that the velocity is 100 ft. per sec. and the diameter is 30 in. (b) Compare the results obtained for the four cases.

38. (a) Explain what is meant by the mean hydraulic radius. (b) Discuss the use of this radius in flow problems for sections far different from the circular.

39. (a) What is the probable value of the upper limit of the friction factor for turbulent flow in a rough conduit? (b) What evidence supports this value? (c) Is this upper limit the same for viscous flow?

40. (a) The applications of the Unwin-Babcock formula are best suited to what fluids and sizes of pipe? (b) Using this formula, calculate the drop in pressure of steam flowing through 200 ft. of 3-in. standard steel pipe (horizontal and insulated) with an initial velocity of 6000 ft. per min., if the initial pressure is 150 lb. per sq. in. abs. and the initial temperature is 600 deg. fahr. (c) Compare the result in (b) with that obtained by using Eq. (609). Ans. (b) 4.91 per sq. in. (c) 4.05 lb. per sq. in.

41. (a) What are suitable velocities to use in steam mains for reciprocating engines? (b) For turbines? (c) For heating mains? (d) Explain fully why such a wide range is involved.

42. What are suitable velocities to use for various kinds of pipe lines carrying water?

43. An insulated heating main is 1 mile long and made up of 10-in. standard new steel pipe having a thickness of 0.365 in. The steam enters this main with a pressure of 200 lb. per sq. in. abs. and a temperature of 500 deg. fahr. (a) If the velocity at entrance is 6000 ft. per min. find the rate of flow in pound per hour. (b) Calculate the pressure drop by Eq. (609), using the initial velocity. (c) Using the result from (b), calculate the average velocity in the line and then determine Δp by Eq. (609). (d) Calculate the drop in pressure by Eq. (591b), using the average temperature obtained by the aid of the results in (b) and assuming an average value of R to be 82. (e) Discuss these results. Ans. (a) 72,500. (b) 36.2 lb. per sq. in. (c) 40.6 lb. per sq. in. (d) 40.7 lb. per sq. in.

44. The products of combustion are discharged from an air preheater at the rate of 109,000 lb. per hr. and at a temperature of 300 deg. fahr. If these gases at a pressure of 14.7 lb. per sq. in. abs. now pass through a light riveted sheet-metal duct 72 in. in diameter and 200 ft. long, find the drop in pressure in this duct assuming the gas constant R to be 52 and that the absolute viscosity is the same as that of nitrogen.

Ans. 0.034 in. of water.

45. (a) Why is it desirable to have standard specifications for pipe and pipe fittings? (b) How is the size of pipe specified? (c) When is "standard weight" pipe used? When is "extra strong" used? (d) What kind of pipe would be used for high-temperature service? Why?

46. (a) In a steam power plant what pipe lines are needed? (b) Name and discuss the basic requirements for piping systems. (c) In multiple-unit plants what systems of pipe arrangements are used?

47. Make a sketch of and discuss (a) the single-header system of piping, (b) the double-header system, (c) the ring system, and (d) the unit system.

48. (a) What are the chief materials used in power-plant pipe lines? (b) Discuss the limitations of each.

49. Find the minimum thickness of a 6-in. lap-welded steel pipe used as a carrier of superheated steam at a pressure of 450 lb. per sq. in. abs. and a temperature of 600 deg. fahr.

Ans. 0.31 in.

50. (a) Why is the thermal expansion of pipe lines an important problem? (b) What are some of the means used to provide for this expansion?

51. (a) In what ways may two lengths of pipe be connected? (b) Make a sketch of and discuss the various forms of flanges.

52. (a) What materials are used for gaskets, and for what field is each adapted? (b) What are the advantages of welded joints?

53. (a) Under what conditions should a globe valve be used? An angle valve?

A gate valve? (b) Of what materials are valve bodies made? (c) What requirements must be met by the materials used in valve stems and discs? (d) Why is the pressure on the two sides of a large high-pressure valve equalized before opening it?

CHAPTER XLIII

1. (a) Define the term "draft" as it is used throughout this chapter. (b) What elements may make up the draft system of a steam-generating plant?

2. From what sources may the energy be obtained to produce the required rate of flow through a draft system? How much energy must be supplied in any case?

3. Classify draft systems from the standpoint of the kind and location of equipment used to produce the draft.

4. By means of a suitable diagram, explain how the *static draft* of a chimney is produced.

5. What is meant by the term "stack effect"? To what extent does the stack effect depend upon a reduction in the temperature of the stack gases while they are *rising* in a chimney? Why?

6. What are the chief elements which cause resistances to the flow of air and gases in the draft system of a steam-generating unit? What is the meaning of the term "resistance" as applied here?

7. At what point in a natural-draft system will the draft be a maximum, (a) during normal operation, and (b) if the flow of air and gases is completely shut off by a choked fuel bed? Why are these two points not usually at the same elevation?

8. Why must the total draft in a natural-draft system be somewhat larger than the sum of the draft losses due to the resistances in the system?

9. A steam-generating unit having a natural-draft system similar to that shown in Fig. 868 is to be operated at a combustion rate of 22 lb. per hr. per sq. ft. of grate area, using anthracite buckwheat No. 1. The draft required to accelerate the gases and to overcome the resistance within the chimney may be taken as 0.2 in. of water. (a) If the temperature of the ambient atmosphere is 60 deg. fahr. and the mean temperature of the stack gases is 450 deg. fahr., find the minimum allowable chimney height for a plant at sea level. (b) How high must the chimney be for the same operating conditions if the plant is located at an altitude of 2000 ft. where the normal barometric pressure is 13.66 lb. per sq. in.? Ans. (a) 257 ft. (b) 275 ft.

10. A steam-generating plant at sea level has a chimney 225 ft. high, serving a natural-draft system identical with that shown in Fig. 868, and operating under the same conditions, with an atmospheric temperature of 50 deg. fahr. and an average gas temperature within the chimney of 500 deg. fahr. The addition of an economizer would increase the draft loss between points 4 and 5 by 0.55 in. of water. Will the chimney continue to serve the plant satisfactorily if the rate of combustion is to remain unchanged? Why?

11. Why does the static force of an unbalanced column of liquid, such as that shown at (c) in Fig. 871, truly represent the *velocity pressure* of a mass of gas moving at a uniform speed in a passageway?

12. Explain how and why the location of the draft gages affects the accuracy of draft readings in large steam-generating units.

13. What is the chief advantage of a multiplying manometer? Explain the principle embodied in the inclined-tube draft gage.

14. Why is artificial draft necessary in many plants?
15. Why is a balanced-draft system so named?
16. Why are mechanical-draft fans more extensively used in stationary plants than any other form of artificial draft?
17. What type of fuel-burning equipment requires a forced draft? Why?
18. Which type of centrifugal fan is often preferred for induced-draft installations? Why?
19. How are the size, capacity, and rated capacity of a centrifugal fan expressed?
20. Make cross-sectional sketches of centrifugal fans having (a) forward-curved blades, (b) straight radial blades, and (c) full backward-curved blades. On each sketch show the path of the air through the fan from the inlet to the discharge; also show the directions of the absolute and relative air velocities at the blade tips.
21. For a given peripheral speed and value of the relative air velocity at the blade tip, show how the magnitude and direction of the absolute air velocity at the blade tip change with a progressive change in blade curvature from full forward to full backward.
22. Why does a fan having full backward-curved blades give a higher static efficiency than any other type?
23. If a certain shape of fan blade gives a low ratio of static pressure to velocity pressure, how may this ratio be increased? Where can the increase be most efficiently accomplished? Why?
24. Three fan wheels similar to those shown in Fig. 878 are operated at a peripheral speed of 150 ft. per sec. In each case the relative velocity of the air leaving the blade tip is 100 ft. per sec. For both the forward-curved blades and the full backward-curved blades the angle between R and a radial line is 30 deg. (a) Find the magnitude of the absolute velocity, v , for each wheel, and its angle with u . (b) Find the value of the ratio of the radial and tangential components of v for each wheel.

Ans. (a) 218 ft. per sec., 23.5°; 180 ft. per sec., 34°; 132 ft. per sec., 41°. (b) 0.435, 0.667, 0.87.
25. A forced-draft fan is required to deliver 20,000 cu. ft. of preheated air per min. at a temperature of 120 deg. fahr. and to produce a static gage pressure of 6 in. of water at the discharge opening. (a) Find the static air horsepower. (b) Find the static air horsepower if the air temperature were 70 deg. fahr. and the same weight of air were handled per minute. (c) Find the total air horsepower for the conditions specified in (b) if the discharge velocity is 3000 ft. per min.

Ans. (a) 18.7 hp. (b) 17.24 hp. (c) 18.87 hp.
26. In terms of what quantities may the characteristics of centrifugal fans be stated? Why should the relations between these quantities be expressed graphically rather than analytically?
27. What factors are of chief importance in influencing the shape of the characteristic curves of fans?
28. A centrifugal fan with full backward-curved blades has characteristic curves identical with those shown in Fig. 879. At the point of maximum total efficiency the shaft horsepower required is 31.4 and the rate of discharge is 25,500 cu. ft. of air per min. at 70 deg. fahr., the inlet pressure being one standard atmosphere, and the outlet static gage pressure 5.5 in. of water. (a) Replot the curves, using as coordi-

nates the absolute values of the various quantities. (b) Plot the curves for total air horsepower and static air horsepower as functions of the quantity discharged per minute.

29. If the speed of the fan specified in problem 28 is 1000 r.p.m., find the approximate values of the discharge rate, the total pressure, and the total air horsepower when the speed is reduced to 500 r.p.m.

Ans. 12,750 c.f.m., 1.45 in. of water (gage), 2.9 hp.

30. What type of fan would you choose for (a) forced-draft service with under-feed stokers, and (b) induced-draft service in a plant burning powdered coal? Give adequate reasons for each choice.

31. For the same rated capacity and static discharge pressure, which type of fan would require a driving motor of the lowest horsepower? Why? Which would require the largest horsepower? Why?

32. Why should an induced-draft fan always be tested with inlet boxes in place?

33. For what type of service is a propeller fan best adapted? Why?

34. Explain the principle of operation of the locomotive draft system.

35. What requirements must be met in order that the necessary draft may be produced in the smokebox of a locomotive?

36. Explain how a favorable relation may be established between high draft and low back pressure in a locomotive draft system. How is this relation influenced by nozzle design?

37. Explain the principle of operation of the Thermix stack.

38. Give the comparative advantages of the three types of self-supported chimneys illustrated.

39. Give all the reasons why chimneys may be required at steam-generating plants.

40. Upon what does the diameter of a natural-draft chimney depend? What is the effect of reducing this diameter? What other change then becomes necessary? Why?

41. Estimate (a) the weight of, and (b) the designed total lateral wind pressure on, the shell of the chimney shown in Fig. 889.

42. A certain steam plant would be satisfactorily served by a natural-draft chimney 175 ft. high and 10 ft. in diameter, or by one 200 ft. high and 9 ft. in diameter. Which would be the more economical chimney? Which dimension of a chimney exerts the greater influence on the cost? Why?

43. (a) Find the approximate recommended cross-sectional area of a duct supplying a stoker with 20,000 cu. ft. of air per min. at a static gage pressure of 5 in. of water, if the air is preheated to a temperature of 200 deg. fahr. (b) Find the value of the ratio of velocity pressure to static pressure for this case.

Ans. (a) 6.21 sq. ft. (b) 0.1043.

44. Why may removal of flue dust constitute a major problem in power plant operation?

45. Explain the principle of operation of each class of apparatus used for removing dust from flue gases.

CHAPTER XLIV

1. (a) What type of condenser is most commonly used in modern central stations? Why? (b) For such a condensing system, what auxiliaries are needed?
2. (a) How is the vacuum in a condenser produced, and how is it maintained? (b) In dealing with condenser pressures, why is it preferable to state the absolute pressure rather than the vacuum? (c) Define the term "standard atmosphere." (d) Why is it essential that the condenser pressure be accurately known?
3. (a) In condensing apparatus, what are the sources of the air entrained with the vapor? (b) Why is it desirable to keep to a minimum the amount of air entering a condenser?
4. The absolute pressure in a steam condenser is 2 in. Hg. The temperature within the condenser shell is 90 deg. fahr. If the steam is dry and saturated, how much air enters with each pound of this steam? Ans. 0.658 lb.
5. For a surface condenser, how much condensing water will be required per pound of steam if the condenser pressure is 1.5 in. Hg abs., and the condensing water is raised from 60 deg. fahr. to within 5 deg. of the temperature of the condensate? Assume that the vapor is dry and saturated and air-free, and that the condensate is undercooled 5 deg. Ans. 48.1 lb.
6. (a) What are the advantages of direct-contact condensers as compared with surface condensers? (b) What are the disadvantages?
7. (a) Sketch a simple barometric condenser and explain its operation. (b) What are its advantages and disadvantages?
8. Sketch a low-level jet condenser and explain its operation.
9. (a) Sketch an ejector condenser and explain its operation. (b) What are its advantages?
10. (a) On what bases may surface condensers be classified? (b) In the selection of a surface condenser, what factors are involved?
11. (a) What are the important items to be considered in designing the steam side of a surface condenser? (b) The same as (a) but for the water side. (c) Give the important structural requirements of a surface condenser.
12. (a) Make a sketch of a double-pass unit-type surface condenser and explain its operation. (b) Under what conditions would condensers have their axes vertical?
13. (a) Why is it so important that the steam be distributed equally to all parts of the tube surface? (b) Discuss the special provision for the distribution of the steam, as shown by Figs. 907, 908, and 909. (c) What provisions are made to insure longitudinal distribution of the steam over the full length of the tubes? (d) Why is it desirable to heat the condensate to exhaust temperature, and how is this accomplished?
14. (a) Of what material are condenser shells made, and why are they ribbed and braced? (b) Discuss condenser tubes as to diameter, thickness, and materials used. (c) Sketch and discuss the various types of condenser tube attachments. (d) Describe the various methods used to prevent the collapse of tube sheets. (e) What provisions are made for the difference in the rate of expansion of the tubes and the shell?
15. (a) Make a sketch of the four usual combinations of condenser connections and supports. (b) Discuss their relative merits.
16. Determine the amount of condensing surface needed, and the weight of condensing water required per hour, for a condenser serving a 60,000-kw. turbine-

generator unit under the following conditions: Actual steam rate of the unit = 9.9 lb. per kw-hr.; condenser pressure = 1.21 in. Hg abs.; quality of the exhaust steam = 90 per cent; condensing water enters the condenser at 55 deg. fahr. and leaves at a temperature of 9 deg. below that of the steam; and the coefficient of heat transmission = 550 B.t.u./hr./sq. ft./deg. fahr. Ans. 57,750 sq. ft. and 28,000,000 lb.

17. (a) Name and discuss the factors which influence the heat transmission in surface condensers. (b) What is the effect on heat transmission of different locations of tubes? (c) Discuss the effect of both load and tube-fouling on heat transmission. (d) Discuss the effect of the condensing water temperature on the absolute exhaust pressure for different loadings.

18. (a) Define the term "cleanliness factor." (b) If you were called upon to determine this factor for a condenser, how would you proceed?

19. How does the amount of surface and the amount of condensing water vary with (a) the size of the unit, and (b) the number of passes?

20. Discuss the determination and the use of the curves given in Fig. 932.

21. (a) What types of pumps are used for handling the condensing water supply? (b) Why is the centrifugal one generally of single-stage design? (c) How do the quantity of water delivered, the pressure head, and the power required vary with (1) changes in speed, and (2) the diameter of the impeller? (d) To prevent a shut-down of the unit, what expedients are used?

22. (a) What is the function of the condensate pump? (b) Why is the water supplied to it under a pressure head? (c) Why are duplicate drives used? (d) What should be the capacity of this pump?

23. (a) What are the advantages of the steam-jet air ejector? (b) Make a sketch of one and explain its operation. (c) Define the terms "inter-condenser" and "after-condenser." (d) What advantage results from using an inter-condenser with an air pump? (e) Under what conditions is the steam-jet air ejector unreliable?

24. (a) Describe the jet ejector type of hydraulic air pump. (b) Describe the hurling type of hydraulic air pump.

25. (a) Define the terms "dry vacuum pump" and "wet vacuum pump." (b) With dry vacuum pumps, what methods are employed to increase the capacity and the vacuum obtainable and to reduce the power for a given size of cylinder and a given speed? Explain each method.

26. (a) With a surface condenser, why is it important that the tubes be kept in good condition? (b) Describe the methods used for removing various kinds of deposits from condenser tubes. (c) Why is chlorination of the condensing water beneficial?

27. (a) How are condenser tube failures classified? (b) Which kind of failure causes the most replacements? (c) How can the designer and the operator minimize the number of tube failures?

28. (a) From what sources may air leakage into the steam space of a surface condenser come? (b) How may air leaks be detected? (c) Describe the methods used to locate the position of the leak.

29. (a) From what sources may water leakage into the steam space of a surface condenser come? (b) Describe the various methods used for determining condenser leakage.

30. Describe the "shock" method used for testing condenser tubes.

CHAPTER XLV

1. (a) What are the functions of a feedwater heater? (b) Discuss the advantages and disadvantages of heating feedwater.

2. (a) Distinguish between open and closed heaters. (b) What other names are associated with heaters, and under what conditions is each class used?

3. (a) State the advantages of the open heater. (b) Discuss the main features of the open heaters shown in Figs. 943 and 944. (c) Discuss (1) the live-steam purifier, (2) the deaerating heater, and (3) the jet type of open heater.

4. (a) Discuss the main features of the closed heaters shown in Figs. 945, 946, and 947. (b) What provision is made for tube expansion? (c) How is scale removed from heater tubes?

5. (a) Name and discuss the factors which affect the transfer of heat in closed heaters. (b) Discuss the film-type heater.

6. A four-pass closed pressure heater has 12 brass tubes in each pass, each tube being $\frac{3}{4}$ -in. O. D. and 30 in. long. With saturated steam at 25 lb. per sq. in. gage, 120 lb. of water per minute are heated from 53 to 200 deg. fahr. Find the coefficient of heat transmission.
Ans. 355 B.t.u./hr./sq. ft./deg. fahr.

7. (a) For what kinds of service are open heaters especially fitted? (b) Same as (a) except for closed heaters. (c) Discuss the application of feedwater heating to steam locomotives. (d) Discuss the steam injector as a feedwater heater.

8. (a) What is the function of an evaporator? (b) What are the advantages obtained by using evaporators?

9. (a) On what bases may evaporators be classified? (b) Describe (1) the submerged-tube type of evaporator, and (2) the film-tube type. (c) What provision is made for expansion of the tubes? (d) How is the scale removed from evaporator tubes?

10. (a) What is meant by the term "effect"? (b) In multiple-effect evaporators, what assumption is usually made in regard to the temperature drop in each effect? (c) How are (1) the economy of the evaporator, (2) the capacity of each evaporator, and (3) the quantity of vapor produced, affected by adding more effects to the evaporator?

11. For what different kinds of service are evaporators used?

12. How may the various types of pumps be classified?

13. Discuss (a) reciprocating pumps, (b) centrifugal pumps, and (c) steam injectors, considered as pumps.

14. (a) What types of pumps are used as boiler feed pumps? (b) What are the advantages and disadvantages of the reciprocating pump as compared with the centrifugal pump?

15. (a) How may the rate of boiler feeding be controlled? (b) What precautions should be taken to see that the boiler has a continuous supply of feedwater? (c) Discuss the factors which influence the allowable suction pressures of boiler feed pumps.

16. (a) What is the function of a steam trap? (b) Classify traps according to the principles on which they operate. (c) Discuss the principle of operation of each class of trap given in (b).

17. (a) What are steam separators? Where are they used? (b) List and discuss the principles upon which they operate.

CHAPTER XLVI

1. Explain fully the necessity for treating make-up boiler feedwater in order that it may be used successfully.
2. Name and define the more important impurities present in natural water, and state the characteristics imparted to the water by them.
3. Explain (a) the meaning and (b) measurement of the *pH* value.
4. Discuss the use of proprietary boiler compounds.
5. Explain how water is clarified. What is the difference between clarification and softening?
6. What dissolved substances produce hardness in water? What is the nature of the hardness in each case?
7. Explain fully the ways in which water may be chemically treated so that the undesirable effects of the various hardness-producing substances may be nullified.
8. Explain the operation of a hot-process softener.
9. Explain the principle of the zeolite process. What are the advantages and disadvantages of this process?
10. What is the purpose of the phosphate treatment? Of what does this treatment consist? What are its advantages?
11. Define *foaming* and *priming*. How are these actions produced? How are they prevented?
12. What is the function of a blowdown system? How is a blowdown cycle (a) developed, (b) controlled, and (c) operated?
13. Explain the continuous blowdown system. How may it be related to the thermal performance of the plant?
14. Why are dissolved gases very undesirable when present in boiler water? Upon what factors does the solubility of a gas in water depend?
15. Explain why the gases released from water are relatively more objectionable in the boiler than is the air originally absorbed outside of the boiler.
16. Explain the various processes used to deaerate water. Which is most useful in the power plant? Why?
17. Explain how deaerators are incorporated into power plant systems, and how they may contribute to good thermal performance of the plant.
18. Describe the appearance of a boiler plate which has been subjected to caustic attack. Why is the result of this attack known as *embrittlement*?
19. Explain fully where and how caustic embrittlement is produced. What metallographic phenomena are involved in this process?
20. Explain how caustic embrittlement may be inhibited, and how the inhibiting agent acts.

CHAPTER XLVII

1. (a) What is a sling psychrometer, and why is it used? (b) Explain why the wet-bulb thermometer of a sling psychrometer attains a steady temperature, between the dry-bulb temperature and the dew point, for any given condition of the atmosphere.
2. Define (a) humidity, (b) relative humidity, and (c) absolute humidity.

3. For a dry-bulb temperature of the atmosphere of 80 deg. fahr., a wet-bulb of 70 deg. fahr., and a barometric pressure of 28.5 in. Hg abs., find (a) the partial pressure of the water vapor, (b) the dew point, (c) the relative humidity, (d) the absolute humidity, (e) the heat content of the mixture per pound of dry air, (f) the volume occupied by the mixture per pound of dry air, and (g) the specific volume of the mixture. Ans. (a) 0.632 in. Hg abs. (b) 65.5 deg. fahr. (c) 0.613. (d) 0.0141 lb. (e) 44.24 B.t.u. (f) 14.6 cu. ft. (g) 14.4 cu. ft. per lb.

4. Outdoor air, with a dry-bulb temperature of 40 deg. fahr., wet-bulb of 35 deg. fahr., and barometric pressure of 29 in. Hg abs., is heated and humidified under steady flow conditions to a final dry-bulb temperature of 70 deg. fahr., and a relative humidity of 40 per cent. (a) Find the weight of water vapor added to each pound of dry air. (b) If the water is supplied at 50 deg. fahr., find the quantity of heat added per pound of dry air. Ans. (a) 0.00314 lb. (b) 10.61 B.t.u.

5. Outdoor air, with a dry-bulb temperature of 95 deg. fahr., a relative humidity of 50 per cent, and a barometric pressure of 29.5 in. Hg abs., is cooled and dehumidified under steady flow conditions until the air becomes "saturated" at 60 deg. fahr. (a) Find the weight of water vapor condensed per pound of dry air. (b) If the condensate is removed at 60 deg. fahr., find the quantity of heat removed per pound of dry air. Ans. (a) 0.0068 lb. (b) 15.88 B.t.u.

6. (a) What is the "adiabatic saturation" process? (b) In such a process, what happens to (1) the dry-bulb temperature, (2) the wet-bulb temperature, (3) the relative humidity, and (4) the absolute humidity?

7. Find, using two methods, the absolute humidity of air having a dry-bulb temperature of 75 deg. fahr., a wet-bulb of 60 deg. fahr., and a total pressure of 27 in. Hg abs. Ans. 0.0087 lb. per lb. of dry air.

8. (a) Why, and by exactly how much, is the "total heat" of a mixture of dry air and water vapor less than the heat content of the same mixture? (b) Prove that the "total heat" of a mixture of dry air and water vapor remains constant during the process of "adiabatic saturation." (c) Does the heat content of the mixture remain constant during such a process? Explain.

CHAPTER XLVIII

1. Explain how water is cooled by air when (a) the water temperature exceeds the dry-bulb temperature of the air; (b) when the water temperature is lower than the dry-bulb but higher than the wet-bulb temperature of the air.

2. What is the "cooling head" when the temperature of the water is 110 deg. fahr. and the wet-bulb temperature of the air is 70 deg. fahr.? Ans. 2.295 in. Hg.

3. If water is cooled from 110 to 85 deg. fahr. by air with a wet-bulb temperature at 70 deg. fahr. find (a) the cooling efficiency according to the A.S.M.E. definition, and (b) the effectiveness of the cooling, using "cooling heads."

Ans. (a) 0.625. (b) 0.72.

4. (a) Describe the arrangement of a cooling pond. (b) Find the approximate area of pond surface required to cool 1,500,000 lb. of water from 115 to 95 deg. fahr. with air having a dry-bulb temperature of 80 deg. fahr. and a relative humidity of 50 per cent. (c) For an air velocity of 440 ft. per min., find the approximate weight of water evaporated per hour in (b). Ans. (b) 116,000 sq. ft. (c) 54,000 lb.

5. Describe the arrangement of a spray pond.

6. Name and describe four different types of cooling towers.

7. A cooling tower is to cool 638 gal. of entering water per minute from 108 to 88 deg. fahr. with air entering the tower with a dry-bulb temperature of 72 deg. fahr., a relative humidity of 60 per cent, and a total pressure of 29.5 in Hg abs. The air leaves the tower at a temperature of 93 deg. fahr. with a relative humidity of 100 per cent. Find (a) the weight of dry air that must flow through the tower per minute, (b) the volume of the air entering the tower per minute, and (c) the water evaporated, in percentage of that entering the tower. Ans. (a) 3430 lb. (b) 47,400 cu. ft. (c) 1.6.

8. A mechanical-draft cooling tower, having 105,000 sq. ft. of effective filling surface (in contact with water), is guaranteed to cool 2000 gal. of water per minute from 105 to 85 deg. fahr. when the wet-bulb temperatures of the air are 65 deg. fahr. at entrance and 85 deg. fahr. at exit, and when the air velocity through the tower is 650 ft. per min. (a) Find the performance factor, F , for the guarantee conditions. (b) If F is proportional to the 0.86 power of air velocity, is the tower meeting the guarantee when it cools 2000 gal. of water per minute from 110 to 92 deg. fahr., provided the wet-bulb temperatures of the air are 70 deg. fahr. at entrance and 95 deg. fahr. at exit, and the air velocity is 550 ft. per min.? Ans. (a) 3.06 B.t.u. per min. per sq. ft. per in. Hg. (b) Yes.

CHAPTER XLIX

1. (a) What general factors must be considered when selecting a site for a new plant? (b) Upon what factors does the design of the plant structure depend? (c) Upon what does the complexity of a plant generally depend? Why? (d) What layouts are generally followed in the arrangement of the equipment in a power plant? Why? (e) In selecting the equipment for a power plant what are the main systems to be considered?

2. What are the most characteristic features of each of the plants illustrated in Figs. 979 to 985 inclusive?

3. (a) Discuss the various hookups used in steam plants operating on the Rankine cycle. (b) What are the main advantages and disadvantages of using the regenerative cycle?

4. (a) Describe the method of draining the heaters used in the plant shown in Fig. 986. (b) How does this system of feedwater heating differ from that shown in Fig. 987?

5. Discuss the feedwater heating system shown in Fig. 988.

6. Discuss the advantages and disadvantages of the various means used to remove the condensate from a closed regenerative feedwater heater.

7. (a) How may the plant auxiliaries be driven? (b) What fraction of the gross power output of a station is likely to be required by the auxiliaries?

8. What are the most important items to be considered in estimating the proper energy balance of an industrial power plant?

9. What analysis should be made before an industrial power plant is built?

10. (a) In what ways is the term "energy balancing" applied in power-plant work? (b) Explain how to make an energy balance for a proposed plant.

11. Explain how the cost per unit of energy delivered by a plant is dependent upon the character of the load.

12. Discuss the load curves given in Figs. 989 and 990.

13. What determines the installed capacity needed in a plant? Why?
14. Define the following terms: (a) demand factor, (b) load factor, (c) diversity factor, and (d) capacity factor.
15. (a) What items constitute the fixed charges on a plant? (b) What is a reasonable allowance for this item?
16. (a) What are the chief items that constitute the first cost of a steam power plant? (b) How does this total cost compare with that of Diesel and hydraulic plants?
17. Discuss briefly the effect of steam pressure on the first cost of a steam power plant.
18. Using the data in Table LXXXIX, determine each item as a percentage of the total cost of the boiler plant.
19. (a) What items make up the production cost of energy from a central station? (b) What is, roughly, the range of this cost for modern steam plants? (c) Compare this cost with the fixed charges and with the total cost of energy at the switchboard. (d) What additional items are involved before the cost to the consumer is obtained?
20. Discuss the data presented in: (a) Table XCIII, (b) Table XCIV, and (c) Table XCV.

CHAPTER L

1. (a) What are the principal factors that affect human comfort as influenced by air environment? (b) What three of these factors must the process of air conditioning simultaneously control?
2. (a) What alterations in the properties of the air in an enclosed space are produced by human occupancy? (b) Which of these alterations are now believed to be chiefly responsible for human discomfort?
3. Explain what is meant by (a) effective temperature, (b) average winter comfort zone, and (c) average summer comfort zone.
4. Find the effective temperature for a dry-bulb temperature of 75 deg. fahr. and a wet-bulb temperature of 60 deg. fahr. for (a) still air, (b) an air velocity of 200 ft. per min.
Ans. (a) 69.2. (b) 66.4.
5. At constant dry-bulb temperature, why is effective temperature influenced by (a) relative humidity, (b) air movement?
6. Find the approximate amount of heat and water vapor added to the atmosphere of a theater by 1800 adults, seated at rest, if the dry-bulb temperature is 70 deg. fahr.
Ans. 540,000 B.t.u. per hr; 180 lb. per hr.
7. (a) State the purpose of ventilation. (b) How may the volume of fresh air supplied to a room per minute per occupant be determined? (c) Explain the Hill synthetic air test and chart.
8. (a) Classify ventilating systems. (b) What is meant by the "neutral zone" in ventilation?
9. What is (a) a direct heating system, (b) an indirect heating system?
10. If the rate of heat transfer from a cast-iron radiator to the room air is proportional to the 1.3 power of the temperature difference, find the rate of heat transfer from a radiator in which the mean temperature of the hot water is 180 deg. fahr., to room air at 75 deg. fahr., if the rated surface of the radiator is 50 sq. ft.
Ans. 7890 B.t.u. per hr.

11. In Fig. 998 (b), if $t_1 = 200$ deg. fahr., $t_2 = 180$ deg. fahr., and $Z = 10$ ft., find the head available for circulation, in feet of water at the mean density.

Ans. 0.0762 ft.

12. (a) Classify steam heating systems into four types. (b) Describe a simple system of each type.

13. Describe a differential seal, and explain why its use may be necessary in a vapor system.

14. Discuss the factors that determine the proper size of pipe to be used in a steam heating system.

15. Describe a gravity circulation warm-air indirect heating system.

16. (a) Describe a fan system of indirect heating. (b) What is meant by the split system?

17. Discuss the factors that influence the selection of the heater and the proportioning of the air ducts in an indirect heating system.

18. What is meant by (a) central heating, (b) district heating?

19. What is (a) a unit cooler, (b) a unit air conditioner?

20. (a) Describe a modified air washer that may be used to heat and humidify the air. (b) What limits the absolute humidity that can be maintained in an enclosure during the winter?

21. During the summer months, how may a low indoor effective temperature be maintained by (a) humidification, (b) dehumidification?

22. State several objections to the "adiabatic saturation" process as a means of maintaining low effective temperature during the summer months.

23. Explain how a dew-point thermostat and a dry-bulb thermostat may be used to control the temperature and the humidity of the air in an enclosure served by air-conditioning apparatus.

24. Describe the use of cooling equipment as a warming machine.

25. Describe a typical installation for residential air conditioning.

26. List the principal factors that determine (a) the maximum heating load, (b) the maximum cooling load.

27. Calculate the overall coefficient of heat transfer U for a single glass window with a thickness of $\frac{1}{12}$ in., if the wind velocity outside is 15 m.p.h.

Ans. 1.07 B.t.u. per hr. per deg. fahr. per sq. ft.

28. Calculate the overall coefficient of heat transfer U for a brick veneer frame wall of the following construction: $\frac{1}{2}$ in. gypsum plaster on $1\frac{1}{2}$ in. corkboard, air space, $\frac{2}{3}\frac{5}{2}$ in. yellow pine sheathing, $\frac{1}{2}$ in. cement mortar, and 4 in. brick. Assume a wind velocity of 15 m.p.h., and treat the air space as two film resistances instead of using the thermal conductivity. Ans. 0.11 B.t.u. per hr. per deg. fahr. per sq. ft.

29. A $2\frac{1}{2}$ ft. by 5 ft., double-hung wooden sash window (weatherstripped), with $\frac{1}{16}$ in. crack and $\frac{3}{4}$ in. clearance, is set in a caked masonry wall; find the total infiltration for a wind velocity of 15 m.p.h.

Ans. 338 cu. ft. per hr.

30. A room, having an outside wall construction with $U = 0.1$, has two windows ($U = 1.1$; size and infiltration as in problem 29) in one outside wall and one in the other. The room is over an unheated basement and below a heated room. There are 250 sq. ft. of outside wall surface (exclusive of windows) and 375 sq. ft. of floor surface ($U = 0.2$). The room is to be maintained at a dry-bulb temperature of 70 deg. fahr. with a relative humidity of 25 per cent when the outside dry-bulb tem-

perature is 0 deg. fahr., relative humidity 100 per cent, and wind velocity 15 m.p.h. Assume a temperature difference between the floor and unheated basement of 35 deg. fahr. Find (a) the quantity of heat transmitted through the window glass, (b) the quantity of heat transmitted through the walls, (c) the quantity of heat transmitted through the floor, (d) the approximate quantity of heat required to warm and humidify the infiltrating air, assuming the water for humidifying to be supplied at 45 deg. fahr. and, (e) the total heating load of this room.

Ans. (a) 2890 B.t.u. per hr. (b) 1750 B.t.u. per hr. (c) 2625 B.t.u. per hr.
(d) 1020 B.t.u. per hr. (e) 8285 B.t.u. per hr.

CHAPTER LI

1. Explain what is meant by mechanical refrigeration.
2. Explain why energy has to be supplied to a refrigerating machine to make it function.

3. Discuss the reversed Carnot cycle with reference to its use in refrigeration.

4. (a) What is meant by the coefficient of performance? (b) Derive the expression for this coefficient in general, and (c) for the Carnot.

5. (a) Discuss the operation of an air refrigerating machine. (b) What are its advantages and disadvantages as compared with machines of other types? (c) Discuss the relative merits of "open" and "closed" systems of air refrigerating machines.

6. In an air refrigerating machine the temperatures are as follows: entering compressor, 30 deg. fahr.; leaving compressor, 272 deg. fahr.; entering engine, 60 deg. fahr.; leaving engine, -112 deg. fahr. Compute (a) the heat absorbed per pound of air, (b) the heat rejected per pound of air, and (c) the coefficient of performance.
Ans. (a) 34.1 B.t.u. per lb. (b) 50.9 B.t.u. per lb. (c) 2.03.

7. Check by another method the result obtained for part (c) in problem 6.

8. (a) Make a diagram showing the necessary parts of a vapor compression plant and describe the processes taking place in each part. (b) Why is an expansion valve used in the vapor cycle and an engine in the air cycle?

9. Distinguish between "dry compression" and "wet compression." What are the advantages of each?

10. (a) Define the term "standard ton of refrigeration" and explain how it is derived. (b) Define the term "standard commercial ton of refrigeration." (c) What are the adopted refrigerant pressures for the standard rating of a refrigerating machine? (d) What is the capacity unit used in rating such a machine? (e) What is the difference between the actual ice-making capacity of a machine and its standard rating?

11. (a) If the temperature of the hot body is 100 deg. fahr., and that of the cold body is 10 deg. fahr., find the horsepower per commercial ton required by the Carnot refrigerating machine. (b) Would the same power be required by an actual machine of large capacity for the same hot and cold bodies? Why?
Ans. (a) 0.903.

12. If a large machine produces a ton of ice by delivering 40 kw-hr. to the electric motor driving this machine, find the coefficient of performance if the ice left the plant at a temperature of 15 deg. fahr. and the water entered the plant at 60 deg. fahr. Consider the specific heat of the ice to be 0.5.
Ans. 2.64.

13. Explain the difference between Eqs. (704) and (706).

14. (a) What properties of a fluid determine its value as a refrigerant? Why? (b) Compare the suitability of H_2O , CO_2 , SO_2 , and NH_3 as refrigerants.

15. Determine by the aid of Fig. 1018 the c.p. of the ideal vapor cycle using freon with an evaporator pressure of 30 and a condenser pressure of 100 lb. per sq. in. abs., assuming that saturated liquid enters the expansion valve, and that saturated vapor enters the compressor. Ans. 5.62.

16. (a) Ammonia vapor has a pressure of 10 lb. per sq. in. abs. and a temperature of 20 deg. fahr. Find by the aid of Plate IV the specific volume, heat content, and entropy of this vapor. (b) Determine its degree of superheat and its mean value of c_p from saturated vapor.

17. (a) If one pound of the vapor of the preceding problem were compressed isentropically until its pressure became 100 lb. per sq. in. abs. find the work done during this process, the final volume, and temperature. (b) Find the value of n such that compression, according to the law $PV^n = \text{a constant}$, until the pressure becomes 100, would give the final state as in (a). (c) Find the amount of work done by the compressor in forcing this vapor from the compressor at constant pressure. (d) If the suction pressure remained constant at 10 lb. per sq. in. abs., calculate the net work of this compressor cycle. (e) Show the areas representing the four quantities of work in this problem.

Ans. (a) 102,300 ft.-lb., 5 cu. ft., 346 deg. fahr. (b) 1.285.

(c) 72,000 ft.-lb. (d) 131,100 ft.-lb.

18. (a) Calculate the rate of heat absorption from ammonia condensed at the rate of 200 lb. per hr. at the constant pressure of 153 lb. per sq. in. abs., the vapor entering the condenser at a temperature of 180 deg. fahr. (b) Find the temperature and heat content of the condensate leaving the condenser. (c) If this liquid were now cooled at constant pressure to a temperature of 50 deg. fahr. find its heat content. (d) If the liquids in the two states determined in (b) and (c), respectively, now have their pressures reduced to 20.34 lb. per sq. in. abs. by passing through a perfectly insulated expansion valve, find the resultant qualities. (e) The difference in qualities as found in (d) produces what variation in the amount of heat absorbed by the ammonia in passing through the evaporator coils after throttling, for a flow of 200 lb. per hr.? Ans. (a) 112,900 B.t.u. per hr. (b) 80 deg. fahr.; 132 B.t.u. per lb. (c) 98 B.t.u. per lb. (d) 18.33 per cent; 12.45 per cent. (e) 6840 B.t.u. per hr.

19. (a) If saturated liquid ammonia at a temperature of 80 deg. fahr. were expanded isentropically to a pressure of 59.74 lb. per sq. in. abs., calculate the resultant quality, specific volume, and heat content. (b) The same as (a) except that the final pressure is 20.34 lb. per sq. in. abs. (c) Show this expansion line on a temperature-entropy diagram, and also show on the same diagram the final state resulting from the same drop in pressure through an insulated expansion valve. (d) Determine the drop in heat content in (b) and show the area representing it. Discuss the significance of this area in refrigeration. Ans. (a) 9.75 per cent, 0.493 cu. ft., 128.9 B.t.u. per lb. (b) 16.47 per cent, 2.204 cu. ft. (d) 10.7 B.t.u. per lb.

20. (a) Make a diagrammatic outline of the essential apparatus involved in the ammonia cycle using the compression system of refrigeration and no precooling of the liquid and no cooling during compression. (b) Draw the pressure-volume and the temperature-entropy diagrams to show the states of the ammonia entering and leaving each piece of apparatus. (c) Show the areas representing the heat transferred to and from the ammonia during the cycle. (d) Show the area representing the net work done by the compressor assuming isentropic compression. (e) If the ammonia enters the compressor with a pressure of 24 lb. per sq. in. abs. and a temperature of 10 deg. fahr. and is discharged at a pressure of 150 lb. per sq. in. abs.,

find the magnitude of the energy terms determined in (c) and (d). (f) Determine the coefficient of performance.

Ans. (e) 489.5 B.t.u. per lb., 610.5 B.t.u. per lb., 121 B.t.u. per lb. (f) 4.05.

21. If the machine in problem 20 is to produce 10 standard tons of refrigeration daily (24 hr.) find: (a) The weight of ammonia that must be circulated per hour. (b) The power delivered to the ammonia by the compressor. (c) The power required to drive the compressor assuming it to be 25 per cent greater than (b). (d) Explain why (c) is larger than (b) in all cases. Ans. (a) 245 lb. (b) 11.65 hp. (c) 14.56 hp.

22. (a) Find the size of the suction pipe required in problem 21 if the average velocity in this line is assumed to be 4000 ft. per min. (b) The same as (a) except for the delivery line, in which the velocity may be assumed as 9000 ft. per min. Ans. (a) 1.5 in. (b) $\frac{3}{8}$ in.

23. (a) Determine the specific volume of freon entering a compressor at state *b* in Fig. 1018. (b) For the cycle shown in this figure what would be the volume flowing to the compressor, in cubic feet per hour, for a " $\frac{1}{2}$ ton" machine. Ans. 166.

24. (a) What is the most important piece of equipment in the refrigerating plant? Why? (b) For what refrigerants is the reciprocating compressor best suited? Why? (c) What types of valves are used with such a compressor and what are their relative advantages? (d) What is the effect of increasing the speed of this type of compressor?

25. (a) What methods are used to control the rate of delivery of a refrigerant from a reciprocating compressor? (b) How are compressor cylinders cooled?

26. What are the advantages of using two expansion valves?

27. (a) Upon what does the successful operation of the rotary compressor for refrigeration depend? (b) For what branch of refrigeration is the centrifugal compressor especially fitted? What are the advantages of this type? (c) What are the advantages of the gear type of compressor?

28. (a) What effect has the refrigerant upon the kind of lubricant used? (b) How is the lubricant removed from the vapor flowing to the condenser? (c) How is the quantity of lubricant fed to the compressor controlled?

29. (a) What is the function of a refrigerant condenser? (b) Describe the submerged coil condenser. What are its advantages and disadvantages? (c) Describe the three types of atmospheric condensers. What are the advantages and disadvantages of this type? (d) Describe the double-pipe condenser. What are its advantages? (e) Describe the shell and tube condenser. (f) How does the rate of heat transmission vary with the temperature of the circulating water?

30. (a) What is a precooling and why is it used? (b) Using Fig. 1027 explain the operation of an accumulator, combined with a precooling.

31. (a) What are the functions of an expansion valve? (b) How are these valves controlled?

32. (a) What is the function of the evaporator? (b) Describe the flash system. (c) Describe the flooded system.

33. (a) How is dry ice made? (b) What are its advantages as compared with ordinary ice? (c) What are the chief applications?

34. (a) Using Fig. 1029 explain the vacuum system of refrigeration. (b) What are its chief applications? (c) What are its advantages?

35. (a) Make a sketch of the apparatus employed in an absorption refrigerating process, and state what properties of the substances used make this process possible.

(b) Describe the operation of the (1) generator, (2) absorber, (3) exchanger, (4) analyzer, and (5) rectifier.

36. Calculate the boiling temperature of aqua ammonia under a pressure of 150 lb. per sq. in. abs. if the weight concentration is (a) 15 per cent and (b) 95 per cent.

37. The same as problem 36, except that the pressure is 15 lb. per sq. in. abs.

38. (a) How is the performance of an absorption machine expressed? (b) In what fields is the absorption system best suited? Why?

39. (a) Describe the adsorption system of refrigeration, as shown in Fig. 1031. (b) What is its principal application?

40. Name the most important applications of mechanical refrigeration.

41. Explain the operation of the Norge rollator.

42. What are the main features of the following household refrigerators: (a) General Electric, (b) Frigidaire, and (c) Electrolux?

43. What was the efficiency of the $\frac{1}{4}$ hp. motor used to drive the household refrigerator for which the test results are given in Table CI.

44. (a) For the results given by the last line in Table CI, determine the brake horsepower per commercial ton of refrigeration. (b) Compare this power with that required by a large machine. (c) Explain why there is so much difference.

INDEX

Absolute humidity, 994
 determination, 996
 viscosity, 789
 of gases, 792, 793
 liquids, 792
 steam, 794
 vapors, 793
 water, 792

Absorber (ammonia), 1127, 1128

Absorption (*see* Air dehumidif.), 1078
 of gases by water, 982
 system (refrig.), 1127-1133,
 1139-1143
 Electrolux, 1139-1143

Accumulator, refrigerant, 1121

Adiabatic cooling process, 998
 saturation process, 998

Adsorption air drying, 1078
 refrig. system, 1133

After condenser, 923, 924
 cooler, refrigerant, 1121

Air (*also see* Atmosphere)
 conditioning, 1055, 1076-1081
 control instruments, 1078
 cooling, 1075, 1077
 dehumidification, 1077-1081
 gels for, 1078
 equipment, 1076, 1078, 1079
 residential, 1080
 heating, 1062-1075
 humidification, 1077
 industrial, 1081
 residential, 1080
 unit, 1076
 washing, 1077
 coolers, 1075
 cooling, 1075, 1077
 by adiabatic sat., 998
 load, 1081
 dehumidification, 1077-1081
 adsorption process 1078
 density at altitudes, 995
 duct(s), 867, 1071-1074
 draft losses in, 867
 drop in pressure, 867, 1073
 elbows, 815
 flow in (*see* Flow of fluids)
 velocities (stoker), 867
 (ventil.), 1073
 effect in condensers, 873
 ejectors, 923-926
 hydraulic, 926
 priming, 925
 steam-jet, 923

Air flow (*see* Flow of fluids; Air ducts)
 horsepower of fan, 851
 humidification, 1077
 infiltration into bldg., 1084
 in water, 981
 removal of, 984
 lanes in condensers, 893
 leakage into condensers, 919, 932
 moisture (*see* Air-vap. mix.; Humidity)
 pressure at altitudes, 995
 effect in condenser, 873
 properties, 990-997 (*also see* Part I)
 pumps, 926-929 (*see* Air-removal)
 dry vacuum, 926-929
 hydraulic, 926
 Leblanc, 926
 multi-stage, 927
 reciprocating, 926-929
 rotative (R. D. V.), 926-929
 steam-jet, 923
 wet vacuum, 929
 refrigeration cycles, 1092-1094
 -removal from condensers, 919
 apparatus, 922-929 (*see* Air
 pumps)
 capacity needed, 919-921
 effect of suction temperature
 on, 921
 compression process, 922
 kinds (*see* Air pumps)
 work done by, 922
 required for ventilation, 1059
 saturated, 921, 990, 999
 solubility in water, 919, 982
 specific heat, 1167, 1168
 temperature at altitudes, 995
 -vapor mixtures, 990-1004
 barometric pressure, 995
 standard, 995
 charts, 921, 993, 1000
 Dalton's law, 990
 definitions, 990-995
 dewpoint, 993
 Ferrel formula, 992
 chart for, 993
 humidity, 994
 partial pressure, 990
 processes of, 997-1000
 adiabatic cooling, 998
 saturation, 998
 genl. energy eq. for, 997
 properties, 995-997
 abs. humidity, 996
 heat content, 996

- Air-vapor properties, relative humidity, 994
 total heat, 1002
 volume, 997
 psychrometric charts, 1000-1004
 ratio (dry air), 920
 temperature effect on, 921
 water vapor in, 920, 990
- Air veloc., stoker ducts, 867
 vent. ducts, 1073
 washing, 1077
- Alkalinity of water, 967
- Altitude, effect on:
 atmospheric conditions, 872, 995
 barometer, 872
 dry air, 995
- Ammonia, absorpt. sys., 1127-1133, 1139
 aqua properties, 1129-1131
 as refrigerant, 1102-1105
 chart, Pl. IV, Appendix
 compressors, 1106-1116
 (*see* Compressors, refrig.)
 condensers, 1115-1120
 (*see* Condensers, refrig.)
 glands, stuffing boxes, 1107
 lubrication with, 1098
 materials with, 1119
 Mollier chart, Pl. III, Appendix
 properties (tables), 1162, 1164
 as refrigerant, 1102-1105
 saturated, 1162
 superheated, 1164
- Amortization, 1044
- Analyzer, 1128, 1141
- Angle valves, 835
- Aqua ammonia, 1129-1131
- Atkinson plant flow diagram, 1032
- Atmosphere, std., 872, 995
 altitude effect on, 872, 995
- Atmospheric condenser, 1116
 cooling towers, 1011
 relief valve, 882
- Auxiliaries, condenser, 914-929
 drives for, 917, 918, 1029, 1030, 1036
 energy account (*see* Part II)
- Babcock formula, 822, 1068, 1069
- Balanced-draft, 838
- Barometric condenser, 880
 pressure, 872, 995
- Bell-and-spigot pipe coupling, 833
- Bends (*also see* Elbows)
 corrugated, 832
 creased, 832
 standard pipe, 832
 welding, 835
- Bernoulli's theorem, 784
- Blowdown, boiler, 979-981
 evaporator, 958
- Blowers (*see* Fans)
- Boiler (*see* Part II)
 blowdown, 979-981
- Boiler caustic embrittlement, 985-987
 compounds, 971
 corrosion, 982
 feedwater (*also see* Boiler water)
 acidity, 967
 alkalinity, 967
 characteristics, 966-969
 conditioning (*see* Boiler water)
 conductivity, electrical, 969
 control, 962 (*also* Part II)
 corrosion from, 982
 definitions, 966-969
 foaming, 978
 gases in, 969, 982
 hardness, 967
 impurities, 966, 988
 make-up (amount), 970
 priming, 979
 properties desired, 970
 pumps, 961-963
 governing, 962
 test methods, 966-967
 troubles from, 985
 foaming, 978
 priming, 979
 water conditioning, 966-989
 (*also see* Boiler feedwater)
 chemical purification, 973-977
 clarification, 972
 compounds, 971
 concentration control, 970
 deaeration, 981-985
 evaporation, 978
 filtration, 972
 lime-soda process, 974
 phosphate treatment, 977
 soda-ash treatment, 974
 softening, 973-978
 sulphate-carbonate ratio, 987
 zeolite treatment, 976
- Breechings, 867
 velocities in, 867
- Buildings, air infiltration into, 1084
 power plant, 1020
- Capacity factor, 1043
 fan, 852
 ice making (melting), 1099
- Carbon dioxide as refrig., 1102-1104
 dry ice, 1124
 in water, 969, 982
 Mollier chart, 1166
 specific heat, 1167, 1168
- Carbonate hardness of water, 967
- Carnot cycle for refrig., 1089-1091
- Caustic embrittlement, 985-987
- Centipoise, 789
- Central heating system, 1074
 stations, 1028-1035
- Centrifugal fans (*see* Fans)
 pumps (*see* Pumps)

- Charts, Ellenwood, 1158
 Mollier (*see* Mollier)
 properties, low press. stm., 874, 991
 psychrometric, 1000
- Chemical-precip. softeners, 975-977
- Chézy formula, 820
- Chimney, chimneys, 838-843, 862-867
 draft, 839-843
 -draft cooling tower, 1012
 friction in, 843
 gas constants, (*R*), 840
 velocities, 866
 Kent's formula for, 867
 linings, 864
 proportions, 866
 radial brick, 863
 reinforced concrete, 863
 steel stacks, 863
 Thermix stacks, 864
 types, 862-864
 velocities in, 866
- Chloride content of water, 968
- Chlorination of cond. water, 930
- Cinder catchers, 868
- Circulating pumps, 915-917
 water (*see* Condensing water)
- Clarification of feedwater, 972
- Cleanliness factor (condensers), 907-909
- Closed heaters (*see* Feedwater heaters)
- Coef. of ht. trans. in ammonia cond., 1119, 1120
 evaporators, 952, 1124
 feedwater heaters, 945
 steam cond., 901-908
- Coef. of performance (refrig.), 1090
 for absorption machine, 1132
 air cycle, 1094
 cooling towers (*see* Perform. factor)
 household refrig., 1144
 refrigerants, various, 1103
 vapor cycle, 1097
- Coefficient, friction (*see* Flow, friction factor)
 viscosity, 789
- Comfort zone, 1057
- Compounds, boiler, 971
- Compressor(s) (refrigerating):
 centrifugal, 1113
 clearance, 1109
 delivery rate control, 1109
 gear, 1114
 lubrication of, 1114
 multi-stage, 1110
 reciprocating, 1107-1112
 speed of, 1108
 rotary, 1112
 water jackets on, 1110
- Condensate heating, 895
 pumps, 918
- Condenser(s), 870-934, 1115-1120
 absolute pressure in, 871
 advantages, disadvantages, 879, 885
- Condenser(s), after, 923
 air effect in, 873
 leakage, 919, 932
 detection, 932
 air-removal apparatus, 921-929
 (*see* Air ejectors; Air pumps; Air-removal apparatus)
 ammonia (*see* Cond., refrig.)
 auxiliaries, 914-929
 barometric type, 880
 cleanliness factor, 907-909
 connections and supports, 897-900
 direct contact, 878-885
 advantages (disadv.), 879
 ejector type, 883
 heat content of exh. steam, 875
 transfer in, 901-908
 hotwell arrangements, 891-895
 inter-and-after, 923, 924
 intermediate, inter-stage, 923, 924
 jet, low level, 881-883
 term. temp. diff., 878
 lanes, air, 893
 steam, 892
 maintenance, 929-934
 performance, 873-875
 log of, 902
 pressure, air effect on, 873
 measurement, 871
 correction of, 872
 properties sat. vapor in, 874
 pumping head, 915
 radial flow, 893
 ratio of vapor to dry air, 920
 water to steam, 876-878
 refrigerant, 1115-1120
 atmospheric, 1116
 bleeder, 1116
 counterflow, 1116
 double pipe, 1117
 flooded, 1117
 function of, 1115
 heat trans. rates in, 1119, 1120
 multi-shell, 1118
 shell and tube, 1118
 submerged coil, 1116
 selection of, 886
 surface type, 885-914
 advantages (disadv.), 885
 air lanes in, 893
 arrangements, 887-895, 910
 as a deaerator, 984
 classification, 885
 cleanliness factor, 907-909
 determination of, 908
 condensate heating, 895
 correction factors for test, 906
 desiderata, 886
 design and test standards, 906-909
 economic size, 911-914
 energy equation, 901
 expansion joints, 898

- Condenser(s):
 surface type (*cont.*)
 friction losses (water), 909
 heat transmission in, 900-909
 (*also see* Part II)
 commercial trans. rates, 907
 high vacuum, 885
 horizontal, 885
 individual design, 886, 889
 leakage, 919, 932, 933
 low vacuum, 885, 888
 multi-pass, 885
 practice, recent, 910
 radial flow, 893
 shells, 895
 single-pass, 885
 size, 886
 standards, 906-909
 steam distribution in, 892-895
 structural requirements, 887
 supports, 897-900
 surface required, 901, 910-914
 terminal temp. diff., 878
 tube arrangements, 891-894
 cleaning, 929
 expansion of, 897
 failures, 930-932
 joints, 896
 materials, 895
 sheets, 896
 sizes, commercial, 940
 usual, 895
 spacing, 893
 testing, 934
 types of, 887-894
 unit type, 886, 887
 vacuum gage readings, 872
 vertical, 885
 water distribution, 895
 velocities, 910, 913
 to steam ratio, 877
 wet vacuum pumps for, 929
- Condensing water:
 amount needed, 876-878, 915
 chlorination of, 930
 energy absorbed by, 878
 measurement of, 878
 pumps, 915-917
 amt. of water handled, 915
 character. curves, 916
 drives for, 917
 effect of changes in:
 impeller diam., 916
 rotative speed, 916
 efficiency of, 916
 energy consumption, 916
 pumping head, 915
 velocity of, 910, 913
- Conductivity of bldg. materials, 1083
 (*also see* Part II)
- Connected load defined, 1043
- Constants (*R*), chimney gases, 840
- Conversion factors (viscosity), 791
- Cooling of air, 1075, 1077
 water by air, 1005-1019
 efficiency, 1006
 head, 1006
 ponds, 1007-1010
 towers, 1010-1017
 design, 1015-1017
 energy balance, 1013-1015
 filling, 1010
 perform. factor, 1016
 test, 1018
 selection, 1017
 types, 1011-1013
 atmospheric, 1011
 chimney-draft, 1012
 combination, 1013
 mechan.-draft, 1012
 natural draft, 1011
- Corrosion by water, 981, 982
- Corrugated expansion joints, 833, 898
 pipe bends, 832
- Cost of energy, 1038, 1051-1054
 plant equipment, 1045-1050
 power plants, 1044-1053
 steam, industrial, 1038
- Costs, operating, 1051-1054
- Creased pipe bends, 832
- Critical number, Reynolds', 805
- Cyclone separator, 868
- Dalton's law, 990
- D'Arcy formula, 820
- Deactivator, 984
- Deaerating feedw. heater, 937, 938, 984, 985
- Deaeration of feedwater, 981-985
 by evaporators, 949
 surface condensers, 895, 984
 degree required, 985
- Deaerators, 984
- Degasification (*see* Deaeration)
- Degree of roughness, 816
- Dehumidification of air, 1077-1081
- Demand factor defined, 1043
- Dense-air refrigeration system, 1092
- Density, defined, 790
 (*also see* Parts I and II)
 of air, 995
 altitude effect, 995
 air-vapor mixture (1/vol.), 997
 ammonia (1/vol.), 1162-1165
 gases (*see* Part I)
 mercury, 872
 refrigerants, 1105
 steam (1/vol.), 1150-1157
 low press., 874
 water, 844, 1065, 1150
- Depreciation, 1044
- Dewpoint, 993
 thermostat, 1078
- Dichlorodifluoromethane, 1104-1106
- Dichloromethane, 1105
- Differential seal, 1067, 1068
- Direct heating systems, 1062-1069

- Dissolved oxygen in water, 969, 982
 corrosion from, 982
 solids in water, 966
- District heating, 1074
- Diversity factor defined, 1043
- Draft, 837-869
 artificial, 838
 balanced, 838
 chimney, 838-843
 definitions of, 837
 dust removal from system, 868
 fans, 849-860 (*see* Fans)
 forced, 838
 function of d. apparatus, 838
 gage, inclined tube, 848
 U-tube manometers, 844-848
 induced, 838
 locomotive, 860-862
 apparatus, 861
 maximum, 862
 loss, 841-843
 breaching, 867
 chimney, 843
 ducts, 867
 locomotive, 862
 steam-gen. unit, 841-843
 measurement, 843-849
 mechanical, 838
 fans for, 849-860
 natural, 838, 839
 pressure equivalent, 844
 requirement, 842
 static, 839
 system, 837
 theoretical, 939
 velocity, press. calc., 847
 in ducts, 867
 chimneys, 866
- Drives for auxiliaries, 917, 918, 1036
- Dry air, altitude effect on, 995
 bulb temperature, 991
 compression (refrig.), 1097
 advantages of, 1097
 ice, 1124
 vacuum pumps, 926-929
- Dual drives, 883, 917, 1029
- Ducts, 867, 1071-1074
 draft losses in, 867
 elbows, 815, 867-868
 velocities (stoker), 867
 (ventil.), 1073
- Dust collection (flue), 868
- Economic chimney proportions, 866
 condenser surface, 911-914
 vacuum, 913
 velocity (water in cond.), 913
- Economics of power plants, 1041
- Economizer (*see* Part II)
 in absorption process, 1128
- Edwards air pump, 929
- Effective temperature defined, 1057
 comfort zone, 1056, 1057
- Efficiency, cooling, 1006
 of fans, centrifugal, 852
 propeller, 860
 pumps (*see* Centrif.), 916
- Ejector, air, 923-926
 condensers, 883
- Elbows (*also see* Bends; Angle valves)
 for air ducts, 815, 867-868
 resistance of, 811-816, 868
 standards, 827
- Electric drives, 917, 1029, 1030, 1036
- Electrical conductivity of water, 969
- Electrolux refrigerator, 1139-1143
- Ellenwood chart, 1158-1160
- Embrittlement, caustic, 985-987
- Energy balance:
 absorption refrig., 1131
 cooling tower, 1013-1015
 plant analysis, 1039-1041
 system, central sta., 1028-1035
 industrial plants, 1037
 consumption, auxiliaries, 1036
 circ. pumps, 917
 refrig. mach., 1088
 cost of, 1038, 1051-1054
 equation, air-vapor mixtures, 997
 evaporators, 951
 feedwater heaters, 942
 flow of fluids, 782
 surface condensers, 901
 stream, air compressor, 1161
 Diesel engine, 1161
 steam engine, 1161
 turbine, 1161
- Engler viscosimeter, 798
 kinematic viscosity from, 799
- Enthalpy (*see* Heat content)
- Equivalent diameter (sq. duct), 1073
- Essential auxiliaries, 917, 918
- Ethane, 1105
- Ethyl chloride, 1105
- Ethylene, 1105
- Evaporation (*also see* Evaporators):
 production of process steam, 957
 single and multiple effects, 950-956
 capacity of each evaporator, 955
 economic limit, 956
 economy of, 955
 method of feed, 956
 numerical examples, 952-955
 treatment of feedwater by, 978
- Evaporator(s), 948-958
 (*also see* Evaporation)
 adaptability, 956
 advantages, 949
 arrangements, 950
 as deaerators, 949
 blowdown systems for, 958
 classification of, 943
 energy equation for, 951
 functions, refrig., 1123
 steam, 948

- Evaporator(s), heat transfer in, refrig., 1124
 steam, 952
 process steam from, 957
 purpose of, 935
 refrigerant, 1123
 single and multiple effects, 950-956
 tube forms, 951
- Évasé stack, 864
- Exchanger, heat, 935
 refrig., 1128
- Exhaust, heat content of turbine, 875
 valve (turbine), 883
- Expansion of pipe lines, 830-832
 provision for, 832
 tubes (condenser), 896, 897
 joints, condenser, 883, 898
 pipe line, 832
 traps, 964
 valves (refrig.), 1122
- F-12, freon, 1104-1106
- Fan, fans, 849-860
 capacity, 852
 centrifugal, 849-860
 characteristics, 853-860
 backward-curved blades, 857
 characteristic curves, 854
 curved blades, radial tips, 858
 forward-curved blades, 856
 operating, 854
 straight radial blades, 857
 variable speed, 859
 velocity relationships, 853
 efficiency, 852, 856-860
 forced draft, 850, 851, 855
 horsepower, 851
 induced draft, 850, 851, 857-859
 inlet boxes, 858
 movable inlet vanes, 860
 multi-blade, 850
 performance, 856-860
 propeller, 850, 860
 rated capacity, 853
 selection, 855
 size, 852
 steel plate, 849
 types, 849
 variable speed operation, 859
 velocity relation, 853
- Fanning formula, 820
- Feedwater characteristics, 966-969, 988
 definitions, 966-969
 heat exchangers, 935
 heaters, 936-948 (*also see* Feedwater heating)
 adaptability of, 946
 advantages of, 936
 classification, 936
 closed, 939-944, 1031
 deaerating, 937, 984
 drainage, 963, 1035
 energy equa. for, 942
- Feedwater heaters, film type, 942
 heat transfer, 944-946
 heating surf. needed, 946
 locomotive, 947
 open, 936-939, 1032
 deaerat. 937, 938, 984, 985
 jet type, 939
 low pressure, 937, 938
 purposes, 935
 steam-tube type, 939
 struct. features, 942
 tubes, 939-941
 com. sizes, 940
- heating (*see* Feedwater heaters; Evaporators)
 by heat exchangers, 935
 heat-recovery apparatus for, 947
 locomotive, 947
 special equipment for, 947
- impurities, 966-969, 988
 make-up required, 970
 permanent hardness, 967
 properties, 970
 control of, 970
 desired, 970
- pumps, 961-963
 governing, 962
- purifiers and softeners, 971-978, 981-985, 988
 temporary hardness, 967
 test methods, 966-969
 treatment (*see* Boiler water conditioning)
- Ferrel psychrometric formula, 992
- Filling, cooling tower, 1010-1011
- Filter, feedwater, 973
 flue dust, 869
- Fittings, pipe, 827
- Fixed charges defined, 1044
- Flanges, pipe, 834
- Flow diagram, Atkinson plant, 1032
 Powerton plant, 1034
 State Line plant (Part II)
 South Amboy plant, 1033
- of fluids in conduits, 781-823
 any fluid, 782
 Babcock's formula, 822
 Bernoulli's theorem, 784
 Chézy's formula, 820
 coef. of friction (*see* Flow frict. fact.)
 critical velocity, 795
 D'Arcy formula, 820
 diagrams, steam plant, 1032-1034
 empirical formulas, 821
 energy equation, 782
 Fanning formula, 820
 friction factor, 800-811, 816-821
 losses, chimneys, 843
 condensing water, 909
- Fritzsche formula, 822

- Flow of hot gases, 785
 in chimneys, friction, 843
 pipe, turbulent, 805
 viscous, 800
 rough closed conduits, 816
 short and long lines, 785
 rate of volume, 801
 resistance to, 785
 of elbows, 811-816, 868
 Reynolds' number, 795
 through annular sections, 820
 bends, fittings, valves, 811
 special shapes, 819
 tube nests, 821
 transverse through tube nests, 821
 turbulent, 805, 811
 rough pipes, 808, 816-819
 smooth pipes, 806, 808-811
 Unwin formula, 822
 velocities, pipe, 823
 air duct (stoker), 867
 (ventil.), 1073
 ratio avg. to max., 806, 807
 viscosity influence, 795
 values, 789-794
 viscous, 800-805
 Flue dust removal apparatus, 868
 Fluid flow (*see* Flow of fluids)
 Foaming, 978
 Foster's equation, 812
 Freon, 1104-1106
 Mollier chart, 1106
 Friction coef. (*see* Friction factor)
 factor (flow), 800-811, 816-821
 Frigidaire refrigerator, 1137-1139
 Fritzsche formula, 822
- Gage, inclined draft, 848
 U-tube manometers, 844-848
- Gas, gases (*also see* Part I):
 absolute viscosity, 792, 793
 absorption by water, 982
 cleaning (dust), 868
 constant, R (*see* Part I)
 chimney gases, 840
 water vapor, 990, 991
 content of water, det., 969
 maximum, 982
 density (*see* Part I)
 flow (*see* Flow of fluids)
 in feedwater, 981
 kinematic viscosity, 799
 mixtures (*see* Parts I and II)
 air-vapor mix., 990-1004
 properties (*see* Part I)
 solubility of water, 982
 spec. heats, 1167, 1168
 ratio (γ), refrig., 1105
 velocity in chimneys, 866
 viscosity, absolute, 792, 793
 kinematic, 799
- Gaskets for pipe flanges, 834
- Gear compressors, 1114
 General Electric refrig., 1133-1137
 Gate valves, 835
 Gels, dehumidifying by, 1078
 in refrigeration, 1133
 Generator (refrig.), 1128
 Gilbert, 1200-lb. station, 1028
 Gland (*see* Stuffing boxes)
 Glass, radiant energy through, 1086
 Globe valve, 835
 Governing of boiler feeding, 962
 Gravity heating system, 1069
 standard, 790
- Hardness of water, 967
 Hawthorne wks., water cooling, 1010, 1013, 1014
 Heat balance (*see* Energy balance)
 content, air-vapor mixture, 996
 ammonia, 1162-1165
 carbon dioxide, 1162
 exhaust steam, 875
 freon, 1106
 steam, 1150-1157
 low pressure, 875
 sulphur dioxide, 1162
 exchangers, defined, 935
 blowdown, 981
 flow diagrams (*see* Flow diagrams)
 -recovery (feed heating), 947
 specific (*see* Specific heat)
 transfer (*also see* Part II)
 in absorption refrig., 1131
 adiab. sat. process, 998
 cond., refrig., 1119, 1120
 steam, 900-906
 cooling ponds, 1008
 towers, 1014
 evaporators, 952
 feedw. heaters, 944
 indirect heating, 1072
 vacuum refrig., 1126
 water cooling, 1005
 through building walls, 1082
 with radiators, 1063
- Heaters (*see* Feedwater heaters)
 Heating loads, 1081-1086
 surface, condenser, 901, 910-914
 economic, 912-914
 feedwater heaters, 946
 systems, 1062-1075
 air vent, 1066
 central, 1074
 direct, 1062-1069
 district, 1074
 gravity, 1069
 heat trans., indirect, 1072
 hot-blast, 1071
 -water, 1063-1065
 indirect, 1069-1074
 air ducts for, 1072
 heat trans., 1070-1072
 meters for district, 1075

- Heating systems, plenum, 1072
 radiators for, 1063
 single pipe, 1066
 split, 1071
 steam, 1065-1069, 1071
 trunk line, 1072
 two-pipe, air-vent, 1066
 vacuum, 1068
 vapor, 1067
 warm air (indirect), 1069
- Hexane, 1105
- Holland station, 1028
- Holly loop, 965
- Horsepower, air, 851
 fan, 851
- Hot process softeners, water, 975-978
 -water heating systems, 1063-1065
- Hotwell, condenser, 862, 891-895
 pumps (*see* Condensate pumps), 918
- Household refrig., 1134-1144
 Frigidaire, 1137
 Gen. Elec., 1135-1137
 Norge, 1135
 performance, 1144
 Servel Electrolux, 1139-1143
- Human comfort (air environment), 1055
- Humidification of air, 1077
- Humidity, absolute, 994, 996
 determination, 996
 relative, 994
- Hybrid system of units, 796
- Hydraulic air ejectors, 926
 radius, 819
- Hydrogen-ion concentration, 967
- Hygrostat, 1079
- Ice-making, melting, capacity, 1099
- Impellers, fan, 850
 pump, 916
- Impurities in water, 966, 988
- Indirect heat. systems, 1069-1074
 ducts for, 1072
 heat trans. in, 1072
- Induced draft fans, 850, 851, 857-859
- Industrial air conditioning, 1081
 power plants, 1037
- Infiltration of air into buildings, 1084
- Injectors, steam, 959, 960
- Installed capacity defined, 1043
- Inter-condenser, 923, 924
- Intermediate, interstage cond., 923, 924
- Iso-butane, 1105
- Jackets (water) on air pumps, 922
 compr. (refrig.), 1110
- Jet condensers (*see* Condensers)
 pumps, 958
 hydraulic-jet air ejector, 926
 steam-jet air ejectors, 923
 -type feedwater heaters, 939
- Joints, pipe, 833-835
- Justified investment defined, 1044
- Kent's chimney formula, 867
- Kinematic viscosity, defined, 789
 determ., 798
 gases, 799
 steam, 798
 uses, 797
 water, 797
- Lanes (condenser), air, 893
 steam, 892
- Leakage (condenser), air, 919, 932
 water, 932
- Leblanc air pump, 926
- Lengths, fittings equivalent, 812
- Lime-soda process, 974
- Linings in chimneys, 864
- Load curves, power plant, 1042
 factor, defined, 1043
 heating, 1081-1086
- Locomotive draft, 860-862
 heaters (feed), 947
- Log of condenser performance, 902
- Logarithms common, 1146
 Napierian, 1148
 use, 1145
- Loop, Holly, 965
 ring circuit piping, 829
- Low-pressure feedwater heaters, 937, 938
- Lubrication of refrig. comprs., 1098, 1114
- Make-up feedwater required, 970
- Manometers, 844-849
- Mass (pound mass), 790
- Materials used in condensers, 895
 feed heaters, 937, 939
 pipe lines, 829
 refrigeration, 1119
 tubes, 895, 939
- Maximum demand defined, 1042
- Mean hydraulic radius, 819
- Mechanical draft (*see* Draft)
 refrigeration (*see* Refrig.)
- Mercury column equiv. in water, 844
 corrections, 872
 density, 872
- Meters for district heating, 1075
- Methane, 1105
- Methyl chloride, 1105
- Mixtures, air-water vapor, 990, 995-997
- Moisture in air, 994-996
 humidification, 1077
 removal, 1078
- Mollier charts, ammonia, Pl. IV, App.
 carbon dioxide, 1166
 freon, 1106
 steam, Pl. I, Appendix
 sulphur dioxide, 1166
- Moores Park plant, 1027, 1028
- Motor costs, 1050
 drives, 1029-1030, 1036
- Multi-blade fans, 850
- Multiple-effect evap., 950, 952-956

Multi-stage compressor (refrig.), 1110
 ejector (air), 923
 pumps (air), 927

Natural-draft cooling towers, 1011

Non-carbonate hardness, 967, 974

Norge refrigerator, 1135

rollator, 1113

Nozzles (*see* Part II)

spray pond, 1009

Open-type feedwater heaters, 936-939

Operating costs (power pl.), *def.*, 1051
 data, 1051-1053

Overall *coef. ht. trans.* (*see* *Coef.*)

Oxygen in water, 982
 corrosion due to, 982
 removal, 984

pH-value of water, 967, 968

Partial pressures in air vapor mix., 990
 condensers, 873

water vapor in air, 991

determin. of, 991

Peak load defined, 1042

Pentane, 1105

Performance, *coef. of* (refrig.), 1090
 absorp. system, 1132
 air cycle, 1094
 Carnot cycle, 1090
 vapor cycle, 1097,
 1102
 cooling tower, 1015-1018
 test results, 1018
 factor (cooling tower), 1016
 fan, 855-860
 refrigerants, 1103
 refrigeration, 1098
 coef. of perform., 1090, 1102
 household, 1133-1134

Permanent hardness of water, 967

Phosphate treatment of water, 977

Pigott's roughness curves, 818

Pipe, pipes (*also see* Piping)

bends, 832, 835
 bursting stresses, 824
 commercial weight, 824, 825
 connections, 833, 834
 flanges, 827, 834
 welded, 834

corrugated bends, 832
 joints, 833

creased bends, 832
 elbows, 827 (*also see* Pipe bends)
 equiv. length (fittings, etc.), 812
 expansion, 830-832

corrugated bends, 832
 joint, 833

creased bends, 832
 slip sleeve, 832
 standard bends, 832

extra strong, 825

fittings, standards, 827

Pipe, flanges, 827, 834

facings, 834

gaskets for, 834

flow (*see* Flow of fluids)

Foster equation for fittings, 812

friction factors, 800-811, 816-821

gaskets, 834

high pressure, 826, 827

high temp., 826, 827, 830, 831

joints, 832-834

materials, 829

O. D., 824

seamless, standard, 824, 825

sizes, standard, 824-827

standards, 824-827, 830, 833, 834

stresses allowed, 830, 831

bursting, 824

thicknesses, 825, 826, 830

unions, 833

velocities in, 823

welded, standard, 824, 825

welding, 834, 835

bends, 835

flanges, 834

wrought, 824, 825

Piping (*also see* Pipe):

arrangements (systems), 828

A.S.A. Code for press. piping, 836

condenser, 897-900

double header, 829

drainage, 963

expansion, 830-832

heating, 1062-1075

Power Piping Soc. stds., 836

ring system, 829

separators, steam, 965

single header, 829

steam plant, 827-836

chief lines, 828

requirements, 828

systems, heating, 1062-1075

steam plant, 829

traps, 964

underground, 1074

unit system, 829

Plant (*see* Steam power plant)

factor defined, 1043

Plates: I, between 1158, 1159; II, 1158;

1159; III, 1160; IV, between 1160, 1161,

V, 1161; VI, 1166; VII, 1166; VIII, 1167

Plenum system (heating), 1072

Poise, 789

Poiseuille's law, 802

Ponds, cooling, 1007-1009

Pound force, 790

mass, 790

Poundal, 791

Power, cost of, 1038

houses, 1020, 1024-1029

plant economics, 1041

plants (*see* Steam power plant)

Powerton flow diagram, 1034

Prat stack, 864

- Precooler, refrigerant, 1121
 Pressure drops (*see* Flow of fluids)
 chimney friction, 843
 empirical formulas, 821
 graph. sol., 1068, 1069
 in ducts, 867, 1073
 fittings, 811
 measure. (draft), 843-849
 steam line comput., 809
 turbulent flow, 805, 811
 viscous flow, 800-805
 dry air vs. altitude, 995
 effect of air on condenser, 873
 on viscosity, 793
 equivalent vs. temperature, 844
 measurement (cond.), 871-875
 sat. vap. contacting ice, 994
 static, 847
 total, 847
 velocity, 847
 Pressures adopted in refrig., 1099
 Priming and foaming, 978
 Process steam from evaporators, 957
 Propane, 1105
 Propeller fans, 850, 860
 pump, 916
 Properties of air-vapor mixtures, 995
 ammonia, 1102-1104, 1162-1165
 aqua ammonia, 1129-1131
 carbon dioxide, 1102-1104, 1166
 exhaust steam, 874
 freon, 1104-1106
 steam, 1150-1157
 low pressure, 874, 991
 sulphur dioxide, 1102-1104, 1166
 Propylene, 1105
 Psychrometer, 991
 Psychrometric charts, 1000
 determinations, 991
 formula, Ferrel, 992
 Pump, pumps, 915-919, 926-929, 958-963
 air removal, 926-929
 (*also see* Air ejectors)
 boiler feed, 961-963
 booster, 963
 bucket (*see* Pump, plunger)
 centrifugal, 916, 959
 characteristics, 916
 drives, 917
 efficiencies, 916
 turbine, 959
 volute, 959
 circulating (*see* Pump, condens-
 ing)
 condensate, 918
 condensing water, 915-917
 capacity, 915
 drives, 917
 efficiency, 916
 energy consumption, 916
 Pump, pumps, condensing water,
 heads, discharge, 915
 suction, 916
 control, 962
 direct acting, 958
 drainage (heater), 963
 drives, 917
 dry vacuum, 926
 dual drives for, 883, 917
 duplex, 959
 Edwards wet vacuum, 929
 efficiencies, 916
 ejectors, 923-926
 feedwater, 961-963
 governing, 962
 heads, discharge, condensate, 918
 condensing water, 915
 feed, 962
 suction (*see* Pump, suction
 lift)
 hotwell (*see* Pumps, condensate)
 hydraulic (air), 926
 injectors, 959
 inside packed, 959
 jet, hydraulic (air), 926
 steam (air), 923
 Leblanc, 926
 multi-stage, air, 923, 927
 water, 959, 961
 outside packed, 959
 plunger, 959
 propeller, 916
 reciprocating, 958
 removal, air, 926-929
 heater, 963, 1035
 jet cond., 919
 R. D. V. (air), 926
 steam ejectors (air), 959
 suction lift, circ. pump, 916
 condensate pump,
 918
 feed pump, 963
 submergence, 883
 triplex, 959
 turbine, 959
 types, 958
 volute, 959
 wet vacuum, 929
 Purification, feedwater (*see* Boiler w. cond.)
 Purifiers, live steam, 937

 Radial flow condenser, 893
 Radiant absorption through glass, 1086
 Radiators, 1063
 Rankine cycle plants, 1029-1030
 Rectifier, 1128, 1140
 Refrigerants, charts for (*see* Mollier)
 comparison, 1101-1104
 data, 1103
 properties of, 1102-1106,
 1129-1131
 (*also see* Properties)

Refrigerating (*also see* Refrigeration)

- compressor(s), 1106-1115
 - centrifugal, 1113
 - clearance adjustment, 1109
 - delivery rate, 1109
 - gear, 1114
 - lubrication, 1114
 - multi-staging, 1110
 - reciprocating, 1107-1112
 - speeds of, 1108
 - rotary, 1112
 - water jackets on, 1110
- equipment:
 - absorbers, 1127-1128
 - accumulators, 1121
 - after coolers, 1121
 - analyser, 1128, 1141
 - compress. (*see* Refrig. compr.)
 - condensers, 1115-1120
 - function of, 1115
 - heat trans. in, 1119
 - perform. characteristics, 1119
 - types: atmospheric, 1116
 - bleeder, 1116
 - counterflow, 1116
 - double pipe, 1117
 - flooded, 1117
 - multi-shell, 1118
 - shell-and-tube, 1118
 - submerged coil, 1116
 - economizers, 1128
 - evaporators, 1123
 - heat trans. in, 1124
 - exchangers, 1128
 - expansion valves, 1122
 - generators, 1128
 - machines (*see* Refrig. machines)
 - oil separators, 1115
 - precoolers, 1121
 - pumps (aqua ammonia), 1128
 - rectifiers, 1128, 1140
 - refrigerators (*see* Refrigerators)
 - machines (*also see* Refrigerators)
 - absorption, 1127-1133
 - adsorption, 1133
 - capacity of, 1098
- Refrigerating machines:
 - compressors, 1106-1115
 - energy consumption, 1088
 - household (*see* Refrigerators)
 - lubrication, 1114
 - performance, 1090, 1091, 1100
 - coefficient of, 1090
 - household, 1144
 - rating, 1098
 - standard, 1098
- Refrigeration (*also see* Refrigerating), 1087-1144
 - absorption, 1127-1133
 - adopted press., inlet and outlet, 1099
 - adsorption, 1133
 - applications of, 1134

Refrigeration (*cont.*)

- coefficient of performance, defined, 1090
 - with absorption machine, 1132
- air cycle, 1094
- Carnot cycle, 1090
- household refrig., 1144
- refrigerants, 1103
- vapor cycle, 1097
- compression system, 1087-1098
- cycles, absorption, 1127-1133, 1139-1143
 - adsorption, 1133
 - air, 1092-1094
 - Carnot, 1089-1091
 - Electrolux, 1139-1143
 - vapor, 1094-1098
- data regarding refrigerants, 1103
- dry compr., advantages, 1097
 - ice, 1124
- function of, 1087
- gels for, 1133
- rating, 1098
- standard commercial ton, 1098
 - ton, 1098
- subcooling, 1121
- systems, absorption, 1127-1133
 - adsorption, 1133
 - compression, 1087-1098
 - vacuum, 1125-1127
- temperatures involved in, 1088
 - standards adopted, 1099
- ton of, 1098
- wet compress., 1097
 - advantages of, 1098
- Refrigerators, household, 1134-1144
 - Frigidaire, 1137-1139
 - General Elec., 1135-1137
 - Norge, 1135
 - performance, 1143
 - Servel Electrolux, 1139-1143
- Regenerative cycle examples, 1030-1034
 - heater drainage, 1035
- Relative humidity, 994
 - viscosity, 790
- Residential air conditioning, 1080
- Reynolds' number, 795
 - critical, 805
- Rollator, Norge, 1113
- Room coolers, 1075
- Rotary compressors, refrig., 1112, 1135
 - pumps, air, 926
- Rough pipes, friction factors, 806-808
- Roughness, degree of, 816
- Saint Johns plant, 1026, 1028
- Saturation press. water vap. contacting ice, 994
- Saybolt viscosimeter, 798
 - kinematic visc. from, 799
- Scrubbers, flue gas, 869
- Sedimentation of water, 972
- Separators, cyclone, 868
 - oil, 1115

- Separators, steam, 965
 Silica gel, 1078, 1133
 Single pipe heating system, 1066
 Slug, 790
 Soda-ash treatment, 974
 Sodium phosphate treatment, 977
 Softeners, water, 975-978
 Softening of water, 973-978
 Solubility, gas in water, 982
 South Amboy flow diagram, 1033
 Specific heats (*see* Part I):
 gases, 1167, 1168
 ratio (γ), refrigerants, 1105
 water, 943
 Split heater system, 1071
 Spray heads, 1009
 ponds, 1009
 Stacks (*also see* Chimneys):
 steel, 863
 Thermix, 864
 Standard atmosphere, 872, 995
 Standards for:
 condensers (surf.), 906-909
 pipe and fittings, 824-827, 836
 materials, 830
 piping, 824-827, 836
 refrigeration, 1098
 sulphate-carbonate ratio, 987
 tubing, 940
 ventilation, 1059
 Static air horsepower, 851
 draft, 839
 fan efficiency, 852
 pressure, 847
 Steady flow of fluids, 781-789
 Steam, absolute viscosity, 794
 cost of (industrial), 1038
 drives (aux.), 917, 1029, 1036
 ejectors (air) (*see* Air ejectors)
 flow (*see* Flow of fluids)
 heat content, of exhaust, 875
 (*see* Steam properties)
 heating system, 1065-1069, 1071
 size of pipe for, 1068
 injectors, 959, 960
 jet air pump, 923
 jets for draft, 838
 kinematic viscosity, 798
 lanes in condensers, 892
 line drops, sample calculations, 809
 loop, 964
 piping, 824-836
 power plants, 1020-1054
 arrangements, 1022
 Atkinson plant flow diag., 1032
 auxil. drives, 883, 917, 918, 1029,
 1030, 1036
 complexity, degrees of, 1022
 condensate drainage, 1035
 conservation (energy):
 apparatus for, 1028-1035
 industrial, 1037
 costs, 1044-1053
 Steam, power plants (*cont.*)
 definitions, 1042
 economics, 1041
 energy balance systems, 1037
 analysis of, 1039-1041
 cost, 1051-1054
 equipment, arrangement, 1022
 cost, 1045-1050
 selection, 1023
 systems, 1028
 flow diagrams, 1032-1034
 Gilbert station, 1028
 heater system, 1031-1034
 drainage, 1035
 Holland station, 1028
 load curves, definitions, 1043
 location and design, 1020
 Moores Park station, 1027
 operating costs, 1051-1054
 pipe lines, chief, 828
 (*see* Steam piping)
 Point Breeze plant, 1021
 Powerton station flow diag., 1034
 reciprocating engine plant, 1025
 Saint Johns, N. B., station, 1026
 simple condensing plant, 1024
 South Amboy flow diagram, 1033
 State Line flow diag. (*see* Part II)
 structure, 1020
 turbine, small, 1026-1027
 typical, 1024-1028
 Western Electric, Point Breeze,
 1021
 process, from evaporators, 957
 properties, 1150-1157
 low pressure, 874, 991
 sat. contacting ice, 994
 separators, 965
 traps, 964
 velocities in pipe, 823
 Steel, embrittlement, 985-987
 pipes (*see* Pipes)
 stacks, 863
 Stokers (*see* Part II)
 air duct velocities, 867
 Stresses in piping, 831
 Stuffing boxes, pump packing, 959
 refrig. compr., 1107
 Subcooling (refrig.), 1121
 Sulphate-carbonate content, 968
 ratio, 987
 Sulphur dioxide, as refrigerant, 1102-1103
 Mollier chart, 1166
 Surface condensers (*see* Condensers)
 feedwater heater, 946
 Suspended matter in water, 972
 solids in water, 966
 Sutherland formula, 792
 Tables—A, 1146; B, 1148; C, 1150; D,
 1154; E, 1162; F, 1164; G, 1168; (I to
 XIV in Part I) (XV to LXVII in Part II)
 LXVIII, 796; LXIX, 813; LXX, 825;

- LXXI, 826; LXXII, 831; LXXIII, 840;
 LXXIV, 910; LXXV, 919; LXXVI, 921;
 LXXVII, 931; LXXVIII, 940; LXXIX,
 968; LXXX, 987; LXXXI, 988;
 LXXXII, 994; LXXXIII, 1018;
 LXXXIV, 1045; LXXXV, 1045;
 LXXXVI, 1047; LXXXVII, 1047;
 LXXXVIII, 1048; LXXXIX, 1049;
 XC, 1049; XCI, 1050; XCII, 1051;
 XCIII, 1052; XCIV, 1053; XCV, 1054;
 XCVI, 1054; XCVII, 1053; XCVIII,
 1103; XCIX, 1105; C, 1130; CI, 1144
 Temperature(s), adiabatic sat. (air), 999
 adopted (refrig. std.), 1099
 dewpoint, 993
 difference (terminal), jet cond., 878
 surface cond., 877
 dry air at altitudes (std.), 995
 bulb, 991
 effect on air pump capacity, 921
 -vapor mix. (cond.), 921
 density (water), 844, 1065
 gas solubility of water, 982
 heat trans. coef., *U*, 905
 mercury column, 872
 pH values, 968
 pipe stresses allowed, 827, 831
 sp. heat of gases, 1167, 1168
 water, 943
 viscosities, 792-799
 water column, 844
 softening, 975
 effective (comfort), 1057
 terminal difference, jet cond., 878
 surf. cond., 877
 wet bulb, 991
 Temporary hardness of water, 967
 Theoretical draft, 839
 Thermal conductivity of bldg. mats., 1083
 (also see Part II)
 Thermix stacks, 864
 Thermostat, dew-point, 1078
 Total pressure, 847
 Towers, cooling, 1010-1017
 Traps, steam, 964
 Tri-sodium phosphate treatment, 977
 Trunk line heating system, 1072
 Tube(s), condenser, 895-897, 929-934
 arrangements, 891-895
 cleaning, 929
 expansion, 897
 failures, 930
 prevention, 932
 joints, 896
 materials, 895
 sheets, 896
 sizes, usual, 895
 standards, 940
 testing, 934
 evaporator, 951
 feedwater heater, 940
 expansion and contr., 941
 materials, 939
 Turbine pump, 959
 Turbulent flow (*see* Flow of fluids)
 friction factor, 805
 in pipes, 805
 rough conduits, 816
 Two-pipe air-vent heating system, 1066
U (*see* Coef. of heat trans.)
 U-tube manometers, 844
 Unions, pipe, 833
 Unit air conditioners, 1076
 coolers, 1075
 type condenser, 887
 Units, absolute system, 790
 conversion factors, 791
 hybrid system, 796
 of refrigeration, 1098
 viscosity, 789-791
 Unwin-Babcock formula, 822, 1068, 1069
 Vacuum, breaker, 882
 economic, 913
 gage readings, 872
 corrections for, 872
 heating systems, 1068
 measurement, 871
 pumps (*see* Pumps)
 refrigeration system, 1125-1127
 Valves, angle, gate, globe, 835
 atmospheric relief, 882
 resistance to flow, 812
 Vapor heating system, 1067
 refrigeration cycle, 1094-1098
 Velocities in air ducts, stoker, 867
 ventil., 1073
 chimneys, 866
 flues (breachings), 867
 pipe lines, 823
 Velocity, across pipe section, 801
 average, 801, 802
 maximum, 802
 critical, 795
 head, 783
 pressure, 847
 relations, fan blades, 853
 Ventilating systems, 1061
 Ventilation, 1059-1062
 air required, 1059
 infiltration, 1084
 standards, 1059
 Viscosity, absolute, defined, 789
 gases, 792, 793
 liquids, 792
 steam, 794
 vapors, 793
 water, 792
 coef. of, 789
 conversion factors, 791
 hybrid system of units, 796
 kinematic, defined, 789
 determination, 798
 gases, 799
 steam, 798
 water, 797

- Viscosity, pressure effect on, 793
 relative, 790
 temperature effect on, 792-799
 units, absolute system, 790
 centipoise, 789
 conversion of, 791
 hybrid system, 796
 poise, 789
- Viscous flow, 800-805
 (*see* Flow of fluids)
- Volume of air-water vapor mixture, 997
 rate of flow, 801
 specific (*see* Properties)
- Volute pumps, 959
- Vortex dust collector, 868
- Walls, ht. trans. through bldg., 1082
- Warm air heating (indirect), 1069
- Washer, air, 1077
- Water, absorption of gases by, 982
 air removal from, 981-985
 alkalinity, 967
 carbon dioxide in, 969, 982
 carbonate hardness, 967
 characteristics, 966-969, 988
 chloride content, 968
 chlorination of cond., 930
 clarification, 972
 column corrections, 844
 condensing, 876-879, 915
 (*also see* Condensing water)
 conditioning (*see* Boiler w. cond.)
 chlorination (cond. water), 930
 cooling by air, 1005-1019
 (*see* Cooling of water)
 corrosion by, 981, 982
 deaeration, 981-985
 (*see* Deaeration)
 definitions, 966-969
 density, 844, 1065
 electrical conductivity of, 969
 filtration, 972
 flow in pipes (*see* Flow of fluids)
 measurement (condensing), 878
- Water, gas content of, 969
 gas solubility of, 982
 heat content, sat., 1150
 impurities, 966-969, 988
 jet air pumps, 926
 non-carbonate hardness, 967, 974
 pH value of, 967
 pressure equivalent of an inch,
 844
 purifying (*see* Boiler w. condition)
 sedimentation, 972
 softening, 973-977
 specific heat, 943
 suspended matter in, 972
 testing for impurities, 966-969
 to steam ratio (cond.), 876-878
 treatment (*see* Boiler water cond.)
 vapor and air mixture, 990-1004
 (*see* Air-vapor mixtures)
 constant, *R*, 990
 partial press. determ., 991
 saturated, 990
 sat. press. contacting ice,
 992-994
 superheated, 991
 volume (*see* Steam tables)
 velocity in condensers, 910, 913
 pipes, 823
- Welded flanges, 834
 joints, 834, 835
 pipe, 824-827
- Welding, bends, 835
 types of, 834, 835
- Western Electric Point Breeze plant, 1021
 water cooling, 1010, 1013,
 1014
- Wet bulb depression, 991
 compression (refrig.), 1097, 1098
 vacuum pump, 929
- Window glass, radiant energy through, 1086
 leakage, 1084
- Wrought pipe, commercial, 824, 825
- Zeolite softeners, 976

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